POWER and THE ENGINEER

EVERYTHING ABOUT THE GENERATION AND TRANSMISSION OF POWER

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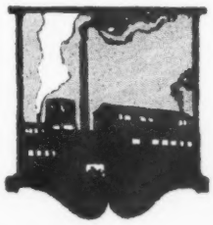

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The Shelton Boiler Explosion

Analysis of Rupture, Starting with Lap Seam Crack. Good Boiler, Good Inspection, Good Care, yet an Explosion

BY F. R. LOW

As briefly mentioned in our issue of December 14, a boiler exploded about 6:30 on the evening of Wednesday, December 1, at the Radcliffe Brothers woolen mill, at Shelton, Conn. Investigation has developed the following particulars:

The boiler was a horizontal tubular, built as was the practice some 20 years ago, with a single under sheet extending for its full length, and three upper sheets. It was 66 inches in diameter, 16 feet long, with one hundred 3-inch tubes, built of ⅜-inch flange steel, stamped 60,000 pounds and with 15/16-inch rivets (driven size) pitched 3¼ inches. It was built 19 years ago by the Bigelow Company, of New Haven.

The Radcliffe mills are operated largely by water power and gas engines, the boiler being used for heating and to supply steam to a small engine. The safety valve had been leaking and had been ground in on Tuesday evening. The boiler was put into commission on Wednesday morning, and ran all day. The mills shut down at 6 o'clock, at which time the night watchman came on duty, and testifies that he found between two and three gages of water and 85 pounds pressure. When he looked at the boiler again some time later the pressure had fallen to 80 pounds, and he had just left the boiler room after his last observation when the boiler let go at 6:29, as indicated by the timekeeper's clock, which was stopped by the explosion. The day fireman, who had not yet banked the fires, was killed; his body was found beneath the overturned front portion of the boiler under the corner, marked with an X in Fig. 6.

The explosion was very violent, as shown by the photographs of the ruins, reproduced herewith, the concussion breaking windows not only in the nearby buildings, but in Derby, across the river.

The sheet portion of the boiler separated into two parts, as shown in Figs. 1 and 4. The initial rupture evidently occurred in the left-hand (as one faced the boiler) longitudinal joint of the rear sheet *A A*, Fig. 4, the fracture of which is an excellent example of the much discussed lap-joint crack. The plate, originally ⅜ inch in thickness, shows an irregular crack, a photograph of which is reproduced in Figs. 2 and 3, extending for the most part two-thirds of the way through the sheet and about 45 inches long. The piece from which the photographs were taken is 29 inches in length, and Fig. 2 shows the edge of the entire

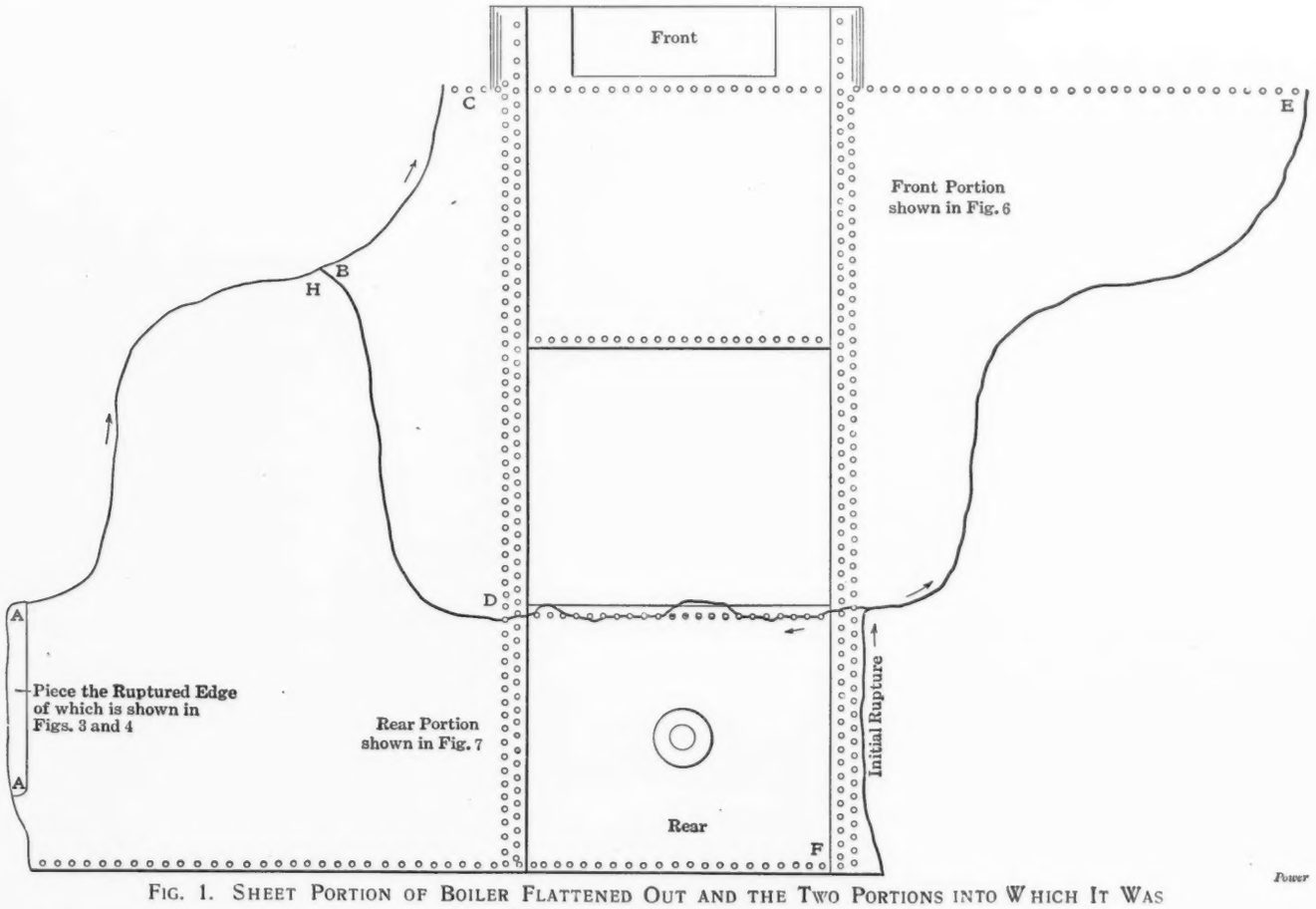


FIG. 1. SHEET PORTION OF BOILER FLATTENED OUT AND THE TWO PORTIONS INTO WHICH IT WAS SEPARATED FITTED TOGETHER

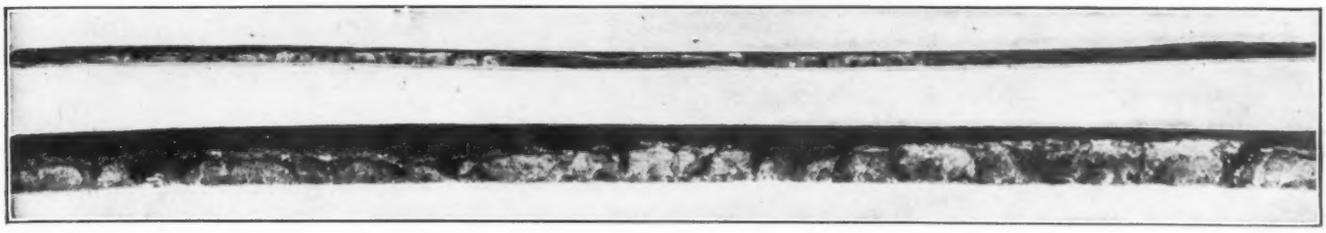


FIG. 2. EDGE OF ENTIRE PIECE
FIG. 3. FULL-SIZED REPRODUCTION OF PORTION OF SAME EDGE

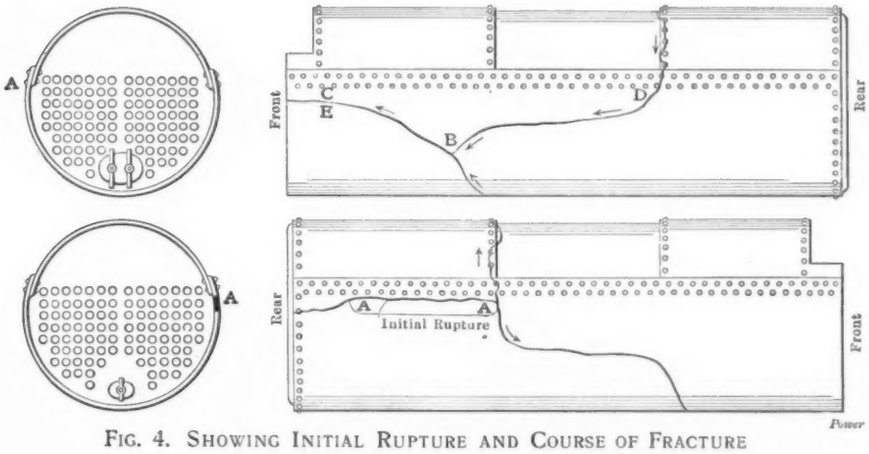


FIG. 4. SHOWING INITIAL RUPTURE AND COURSE OF FRACTURE

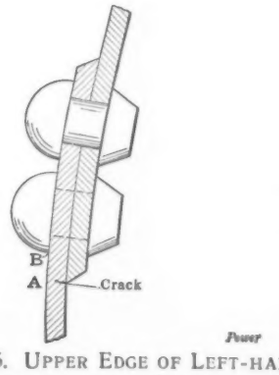


FIG. 5. UPPER EDGE OF LEFT-HAND PLATE SHOWN IN FIGS. 3 AND 4

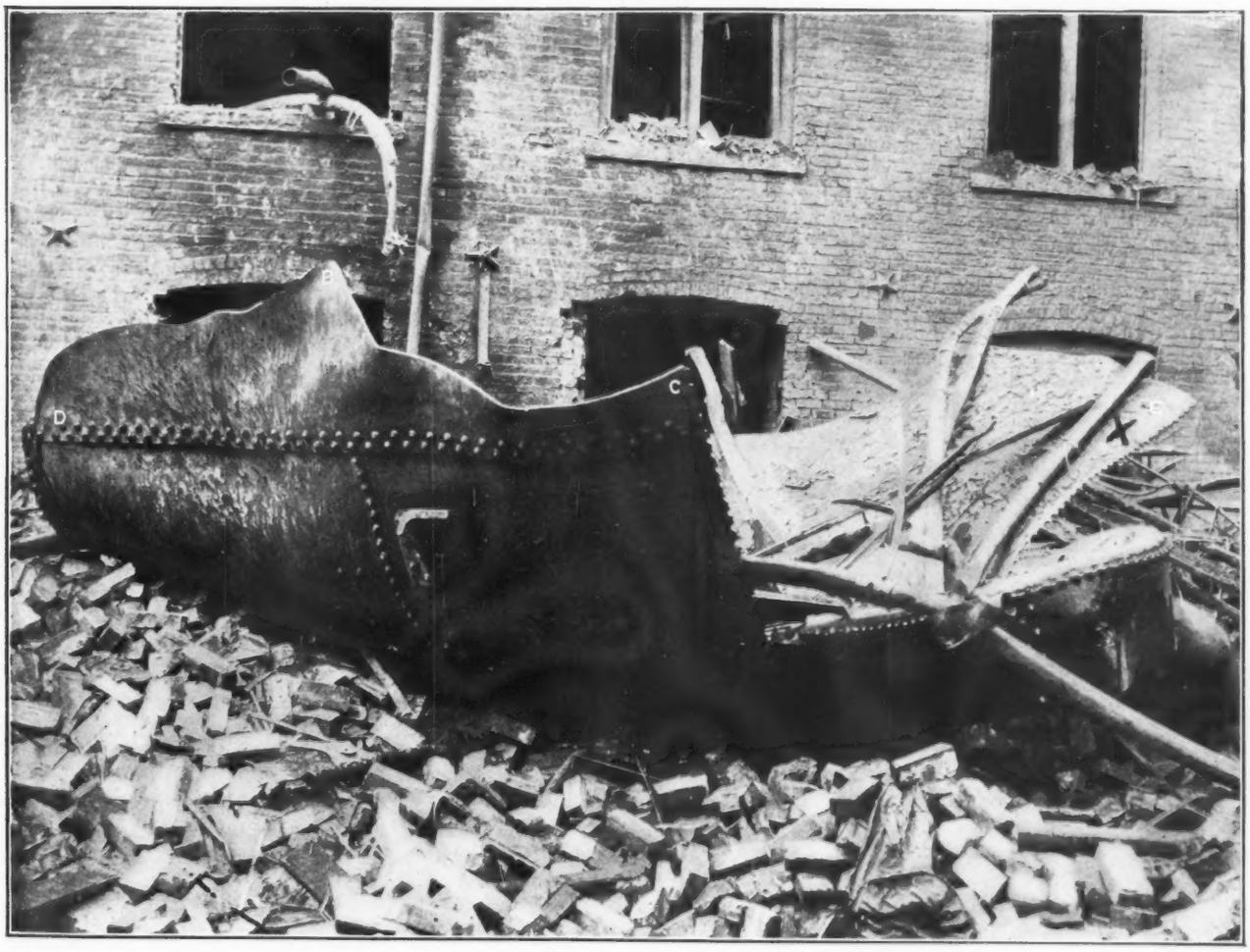


FIG. 6. FRONT PORTION, UNDER WHICH FIREMAN WAS FOUND

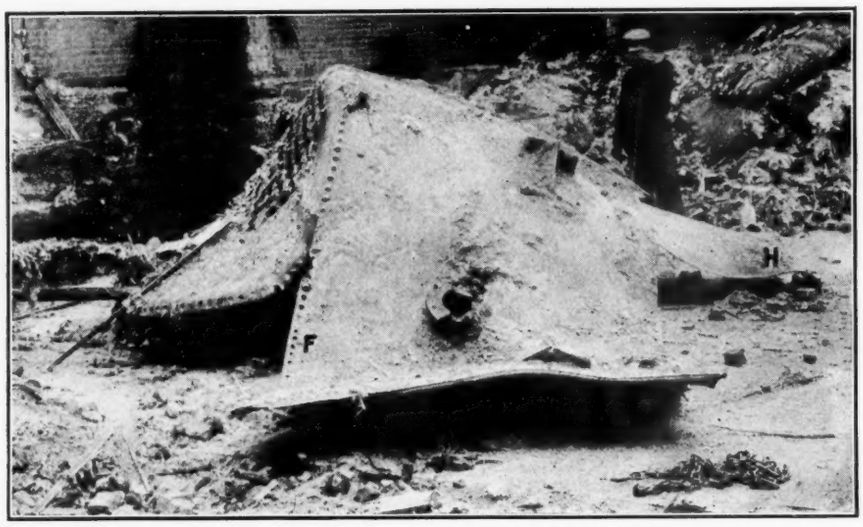


FIG. 7. VIEW OF REAR SECTION OF BOILER

FIG. 8. REMAINS OF POWER PLANT WHICH WAS LOCATED BETWEEN THE TWO BUILDINGS SHOWN

piece, while Fig. 3 is a full-sized reproduction of a portion of the same edge. The fracture apparently ran along this seam and out into the large under sheet, as indicated in Figs. 1 and 4, up and over the girth seam, which failed sometimes in the rivets and sometimes in the sheet; and into the large sheet again at *D*, where it struck the continuous longitudinal point upon the other side, joining the other fracture at *B*, and running out until it struck the rivets of the front head, where it ran almost entirely around the girth seam. The rear portion was thrown backward, and to the right as one faced the boiler, as it would naturally be thrown by the opening of the seam *A A*, while the front portion somersaulted and lay upside down, with its front head, to which the extension front remained attached, near the original location, as shown in Fig. 6. This consists of two of the short upper sheets joined by two fragments of the lower sheet, as will be evident from Fig. 1, where points upon the sheet have been lettered similarly with the same points upon the photograph in Fig. 6. A small piece of the rear head remained attached to the lower sheet.

The under sheet lapped over or outside of the smaller upper sheet, as shown in Fig. 5, and the fracture occurred along the line the position of which is indicated by *A* in that figure. This is below the line of tubes, as is better seen in Fig. 4, so that it was inaccessible for inspection, and even if the inspector could have gotten to it, it was for the most part covered by the upper plate, occurring right along the edge of that plate, as shown in Fig. 5. This is not the calking edge, so that the crack cannot have been induced by the careless use of the calking tool. The position of the piece, the cracked edge of which is shown in Figs. 2 and 3, is indicated in Figs. 1, 4 and 5

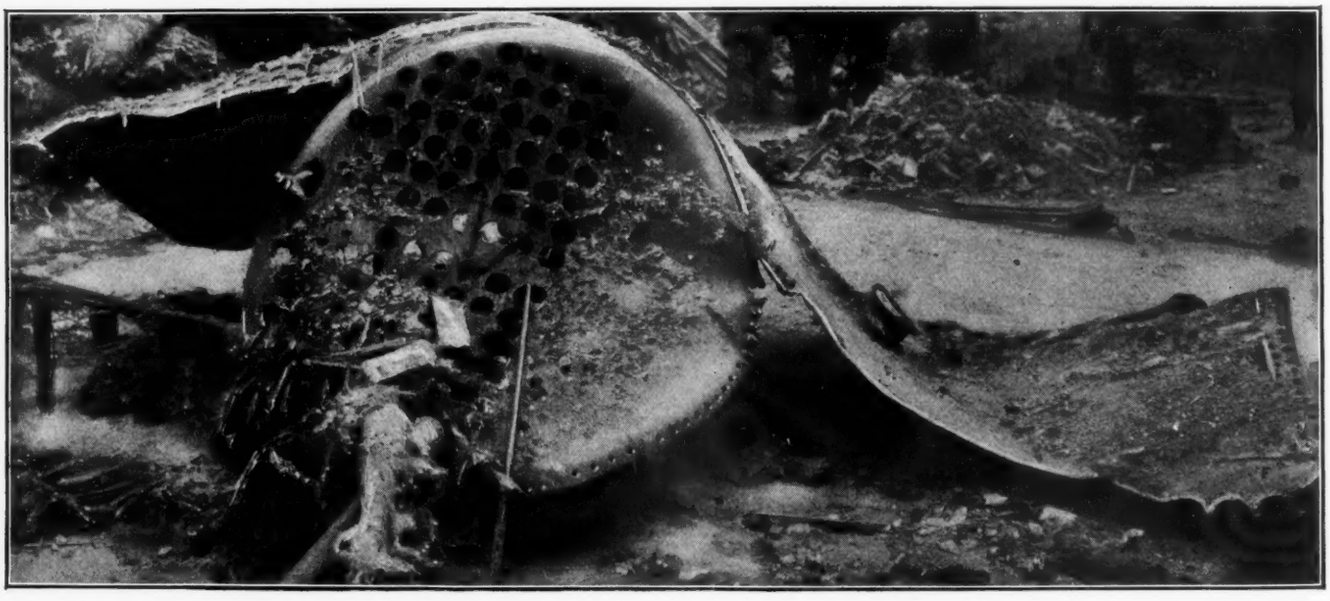
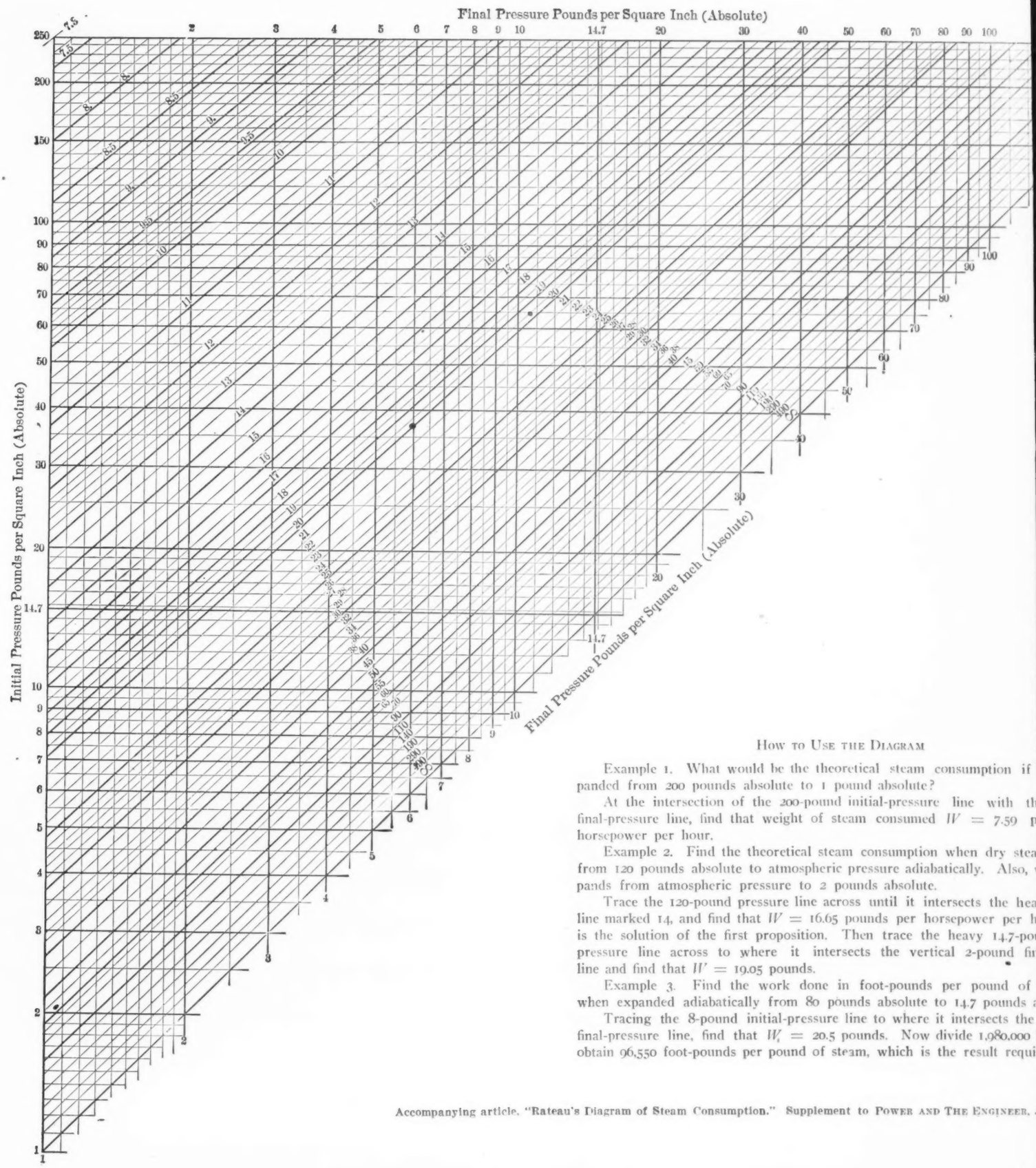


FIG. 9. THE REAR HEAD WITH ATTACHED SHEET


Final Pressure Pounds per Square Inch (Absolute)
Initial Pressure Pounds per Square Inch (Absolute)
Final Pressure Pounds per Square Inch (Absolute)

How to Use the Diagram

Example 1. What would be the theoretical steam consumption if
panded from 200 pounds absolute to 1 pound absolute?

At the intersection of the 200-pound initial-pressure line with th
final-pressure line, find that weight of steam consumed $W = 7.59$ p
horsepower per hour.

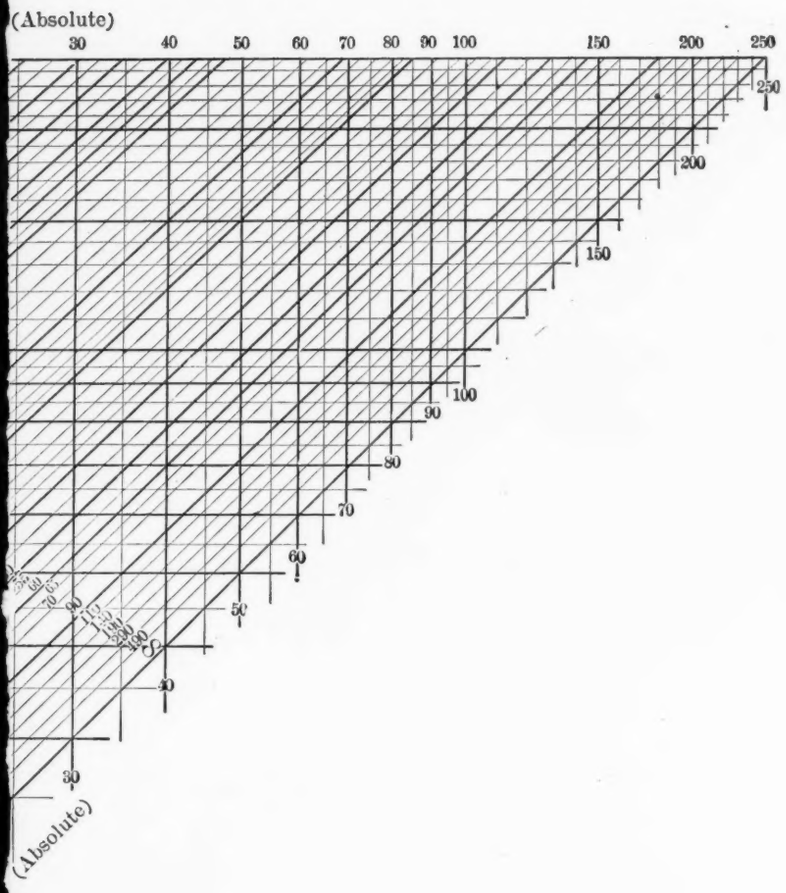
Example 2. Find the theoretical steam consumption when dry stea
from 120 pounds absolute to atmospheric pressure adiabatically. Also, v
pands from atmospheric pressure to 2 pounds absolute.

Trace the 120-pound pressure line across until it intersects the hea
line marked 14, and find that $W = 16.65$ pounds per horsepower per h
is the solution of the first proposition. Then trace the heavy 14.7-pou
pressure line across to where it intersects the vertical 2-pound fin
line and find that $W = 19.05$ pounds.

Example 3. Find the work done in foot-pounds per pound of
when expanded adiabatically from 80 pounds absolute to 14.7 pounds a

Tracing the 8-pound initial-pressure line to where it intersects the
final-pressure line, find that $W_t = 20.5$ pounds. Now divide 1,980,000 b
obtain 96,550 foot-pounds per pound of steam, which is the result requir

Accompanying article, "Rateau's Diagram of Steam Consumption." Supplement to Power and The Engineer, J



How to Use the Diagram

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pound initial-pressure line to where it intersects the 14.7-pound find that $W_1 = 20.5$ pounds. Now divide 1,980,000 by 20.5 and ounds per pound of steam, which is the result required.

onsumption." Supplement to Power and The Engineer, Jan. 4, 1910, Vol. 32, No. 1,

FIG. 10. THE REMAINS FROM ANOTHER POINT OF VIEW

by *A*. The examples of this sort of crack, of which we have previously learned, have been along the edge of the rivet heads, that is, in the line indicated by *B* of Fig. 5, instead of at the edge of the inside sheet.

Figs. 7 and 9 give two views of the rear portion. They would have been more satisfactory if the photographer had used a broom before he did the camera, but referred to Figs. 1 and 2 by the letters which have been added they will help the reader to reconstruct the boiler mentally and follow the course of the fracture, the initial point of which was in the edge facing the observer in Fig. 7.

The violence of the explosion is evidence that the boiler was full of water. The small amount of sound metal left in the cracked sheet precludes any necessity for serious overpressure to produce the rupture, and the riveted seams show no signs of such distress as would have attended the application of anything like the bursting pressure of a sound boiler of this size and thickness. The metal, with the exception of the crack, was in excellent condition and the fractured edges all over the boiler show considerable reduction of area and great ductility. Test pieces have been taken.

The boiler had been inspected externally a couple of weeks before the explosion, and internally last April by an inspector of the Fidelity and Casualty Company, with which company the boiler was insured. The damage is estimated at some $60,000, and the full face of the policy was promptly paid.

The boiler was located in a brick building two stories high, which was also used by the dyeing and bleaching departments of the factory, and which stood between the two high buildings, shown in Fig. 8, which was taken with the camera standing in and looking out of the arch, shown in the rear of Fig. 10. The brick chimney, which stood next to the boiler house, was demolished.

Rateau's Diagram of Steam Consumption

By W. C. Way and H. F. Schmidt

Rateau's diagram for determining the theoretical steam consumption per horsepower per hour has been published elsewhere a number of times, but unfortunately in the shape of practically illegible reprints of a small diagram, in French units, which appeared in the catalog of the Rateau turbine a year or two ago. The diagram on the accompanying insert is a revision of the original, constructed on a scale designed to be of practical value, in which English units have been employed, instead of French.

A peculiarity which presents itself at first glance is that the vertical and horizontal lines are not uniform and there is the same distance between the one- and two-pound ordinates as between the one-hundred- and the two-hundred-pound ordinates. This is because the spacing of the lines is based on the logarithmic scale to the base 10. The reason for the selection of this spacing is that an equation of the form, $\log X = \log Y + \log Z$, etc., when plotted on this paper, will simply result in straight lines. This fact it is well to remember, for when an equation can be reduced to such a form the plotting of the curves it represents is simply a matter of calculating two points and drawing a line through them; whereas, had the ordinary line spacing been employed, it would have been necessary to find at least ten or more points on each curve. Further, in cases where the axes of the curve to be plotted are its asymptotes, or nearly so, in the case of the ordinary rectangular coördinates, it is often difficult or impossible to accurately determine the point of intersection of the curves (especially if there is a series of them), while if plotted on a logarithmic chart they would probably intersect the axes at an angle of about 45 degrees, and consequently be more distinct and susceptible of being read more accurately.

Regarding Rateau's curves, it will be well first to give the formula he derived. This he obtained by a study of the temperature-entropy diagram, from which he obtained an empirical form expressing the theoretical steam consumption in terms of the absolute initial and terminal pressures, which in English units takes the form,

$$W = 1.875 + \frac{16.54 - 1.485 \log P_1}{\log P_1 - \log P_2}.$$

This, it will be seen, is the same type of equation as previously given, and plots as a straight line on a logarithmic chart. In this formula, the pressures are in pounds per square inch absolute, and the steam consumption in pounds of dry steam per horsepower-hour. The expansion, of course, is assumed to be adiabatic, without a drop at admission or exhaust.

Never attempt to calk a leaky seam in a boiler under steam pressure. The jar caused by the hammer blows may cause a rupture of the seam. This caution applies also to steam pipes and valves.

Condensing Plant of Novel Design

Water Turbine on Shaft of Circulating Pump Utilizes Fall of Water on Return to River to Help Pump Fresh Supply

BY FREDERICK RAY

With the impetus given to the condenser business by the increase in power-plant installations and the development of the steam turbine came a more complete knowledge of its adaptability to the various engine installations. Naturally its adoption was slow in rolling-mill practice, where fuel economy seemed to be of minor importance. But as the manifest advantage of removing from the advancing side of the piston a part of the atmospheric pressure even in reversing engines became understood, rolling mills began to adopt condensing systems.

Among the noticeable condenser equipments in American rolling mills is the one at the Saucon plant of the Bethlehem Steel Company, at South Bethlehem, Penn., where the overflow from the hotwells is led to water turbines and its energy used to help operate the pumps which supply the condensers, returning in its descent nearly 30 per cent. of the power expended in raising it to the hight where it is used. As the source of water supply is so far below the condensers, the problem of its elevation and the means by which in returning to the river it is made to give up a part of its energy and really help to pump itself should excite more than a passing interest.

The difficulties surrounding the installation of the condensing equipment for this plant were in themselves not at all unusual for plants located upon the banks of American rivers, but the method of surmounting them seems to be of enough novelty and value to warrant a description.

The location of the plant is on a small plateau on the south bank of the Lehigh river, the ground level being about 62 feet above the mean level of the river. As it was necessary to locate the condensers on this plateau and the circulating water required being about 30,000,000 gallons per day, the pumping end of the condenser equipment became one of considerable magnitude and importance.

Three schemes for this part of the installation were proposed and carefully

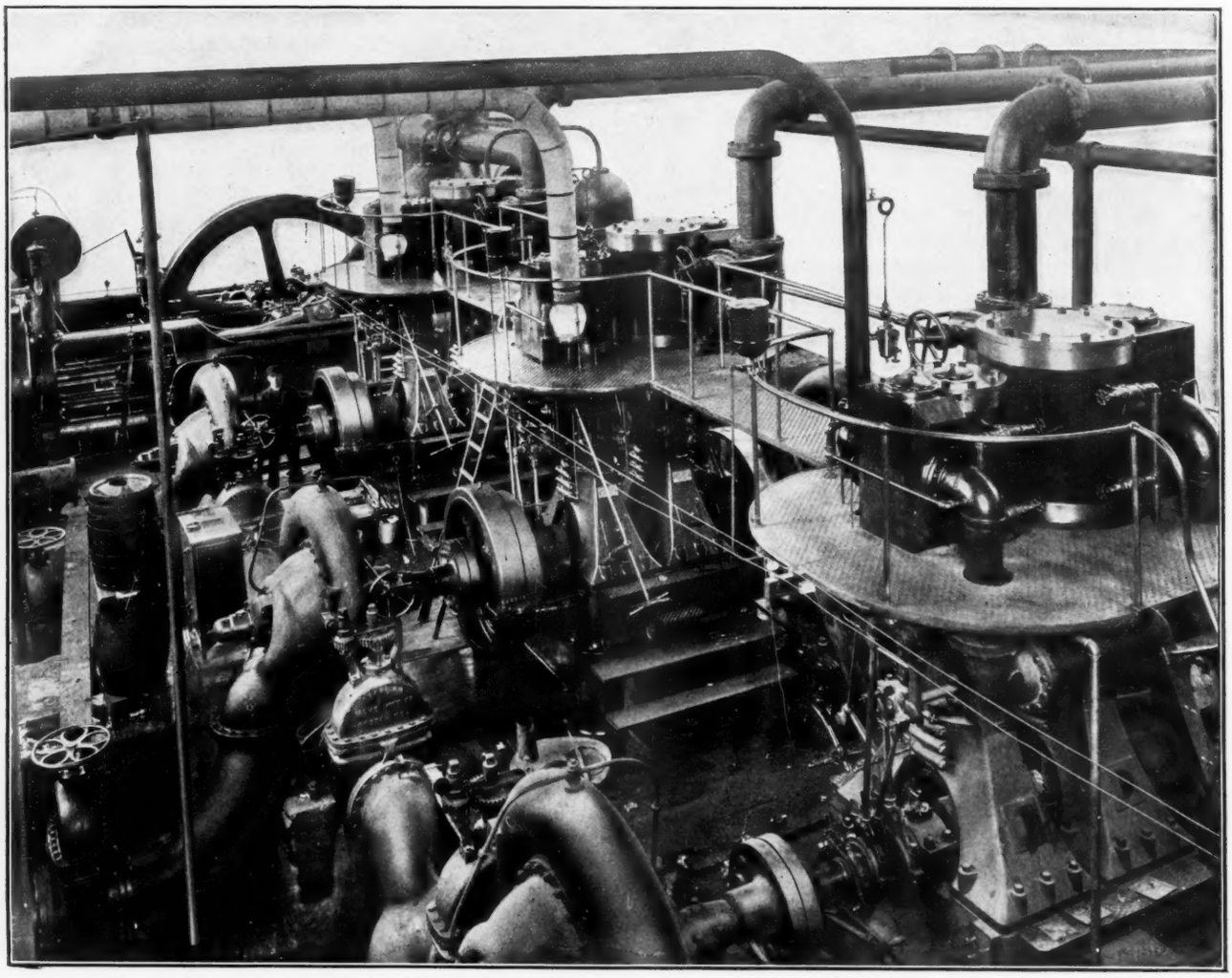


FIG. 1. VIEW OF PUMPING PLANT, SHOWING WATER TURBINES

considered. The first contemplated the use of compound high-duty crank and flywheel pumping engines; the second involved the use of motor-driven centrifugal pumps taking their current from the central power station at the plant, while the third made use of the novel arrangement of engine-driven centrifugal pumps having direct connected to the shaft of each, a water turbine delivering back to the pump a large part of the energy contained in the water on its return to the river. The return-water turbine was also considered in connection with the motor-driven pumps, where it could have been used to equal advantage, but from the fact that it was necessary to locate some general-service high-duty pumps at the river edge, with a boiler plant to drive them, it was found that engine-driven centrifugal pumps making use of this same boiler plant would be equally desirable and of considerably less first cost than the motor-driven pumps with the necessary generating equipment. The high-duty crank and flywheel pumps were found to be much greater in first cost and as there seemed to be no feasible way to utilize the return-water energy, in connection with them, the duty which they were capable of maintaining was not as good as could be obtained with the engine-driven centrifugal pumps in connection with the return-water turbines. The third scheme therefore was the one selected as best meeting the requirements of the situation on the basis of reliability, first cost, cost of operation and repairs.

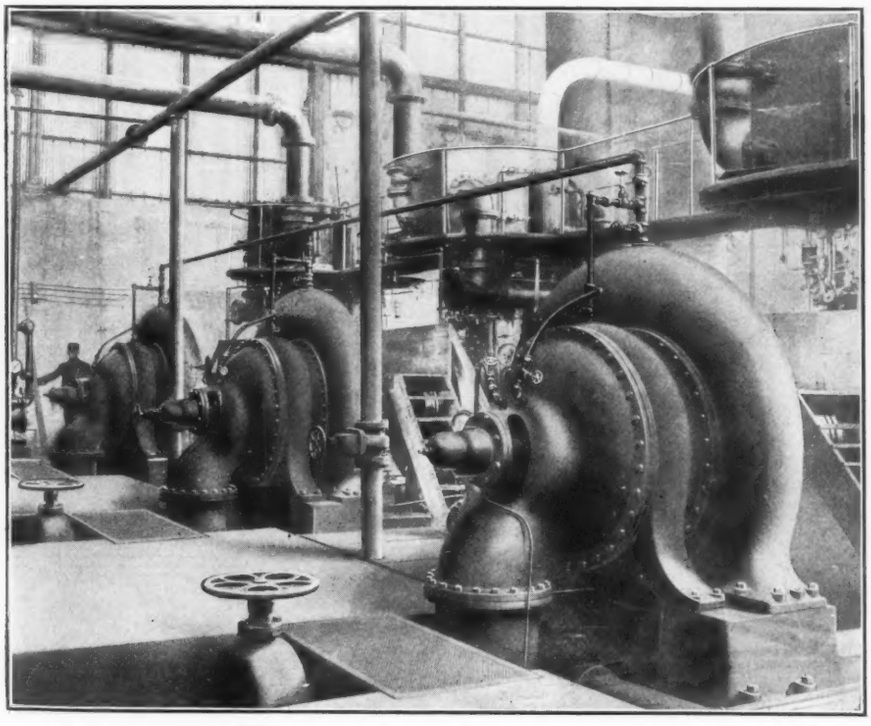


FIG. 2. TWO-STAGE VOLUTE CENTRIFUGAL PUMPS

The Saucon plant considered only as a steel mill has many novel and interesting features which, however, are beyond the range of this article, so that it will suffice to state that the engines for which the condensing equipment was furnished consist of one 40-inch blooming-mill engine, one 28-inch structural-mill engine, and two rail-mill engines exhausting into one condenser, while one 46-inch blooming-mill engine and two Grey mill engines exhaust into another. Each condenser was required to condense 200,000 pounds of steam per hour while maintaining a vacuum of 26 inches of mercury with injection water during the summer months reaching a maximum temperature of 75 degrees Fahrenheit. To meet these conditions two 54-inch barometric condensers were installed at points most convenient to the particular engines served, each condenser including a 12 by 24 and 24 by 24 multiple single-stage rotative dry-vacuum pump. The general arrangement of condenser, vacuum pump and piping is shown diagrammatically in Fig. 4, while Fig. 3 is from a photograph of one of the condensers and gives a good idea of its actual size. The exhaust pipe which is shown rising from the ground to the condenser, is brought from the various engines for a distance of several hundred feet in an open concrete trough, the top of which is at the ground level and covered with removable planking.

Fig. 1 is a view of the engine room from the return-water turbine side of the engines, showing the turbines with water-regulating gate valves, flexible attachment to engine shaft, etc., while Fig. 2 is a view of the pump side of the engine room.

Referring to Fig. 4, it is seen that the circulating water is taken from the river through a series of concrete settling basins, the innermost one of wnich

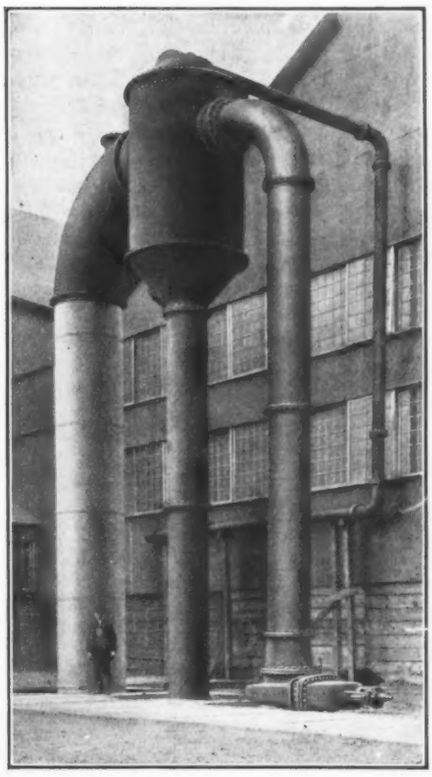


FIG. 3. ONE OF THE 54-INCH CONDENSERS

forms the suction wall of the pumps. From here the water is lifted by suction or flows to the pumps, depending upon the level of the river, and is discharged through two independent lines of 30-inch cast-iron pipe, one leading to each condenser. Passing through the condensers the water overflows from the hotwells through 30-inch cast-iron pipes which discharge into the concrete collecting chamber and weir box. From here the water flows through one 36-inch cast-iron pipe back to the pumping station through the return-water turbines, and into the discharge well located on the land side of the station, emptying into the river below the intake to the settling basins. This 36-inch return-water pipe as it enters the station is connected to a 54-inch standpipe containing a 36-inch inside overflow emptying into the discharge basin.

This standpipe and overflow were provided in the return-water system to afford a discharge for the water in case of intentional or accidental closing of the gates on the water turbines preventing any possibility of raising the level of the water in the hotwells and consequent flooding of the condensers with the ensuing danger of wrecking the vacuum pump or even the engines. The edge of

the overflow pipe is placed enough below the elevation of the condenser-hotwell overflow to allow for the friction in the return-water pipe so that when the maximum normal quantity of water is flowing through the system the level of the water in the standpipe will be at the edge of the overflow when the condenser hotwells are just overflowing freely. The collecting chamber in the return-water system was designed to provide a proper intake to prevent the entrance of air with the return water, and also as a separating chamber in which the air entrapped with the tail water from the condensers would have an opportunity to free itself. At the same time the chamber was designed so as to be adaptable for use as a weir box in connection with any tests that might be desired.

It would have been more economical to have used, instead of two 30-inch discharge pipes, one larger pipe with branches to the two condensers but it is found in practice that such an arrangement will not in general work satisfactorily.

The horizontal distances are so distorted in Fig. 4 that it might be well to state that the collecting chamber is located about 520 feet from the pump house, while the nearest condenser is about 130 feet beyond, and the other 700 feet from the chamber as measured on the pipe lines. The ground line which is shown as dropping down close to the condenser in reality continues nearly level until close to the pump house where it terminates in a steep bluff at the edge of the river.

The pumping equipment is divided into three units, two of which are of sufficient capacity to meet the maximum requirements while the third is held as a spare. Each unit consists of a 20-inch two-stage volute pump direct-connected by means of flexible couplings to a 14¼ and 25 by 14-inch cross-compound vertical Buckeye engine while to the opposite side of each engine is connected, by means of flexible coupling also, a 20-inch return-water turbine, all being mounted upon one cast-iron bedplate. The arrangement of these units is well shown in Fig. 4, while the general arrangement of the three units and the piping is shown in the plan and elevation in Figs. 5 and 6, and by the photographs in Figs. 1 and 2. From these views it is seen that units Nos. 1 and 3 can be connected only to their respective discharge pipes while the middle unit, No. 2, can be connected to either, which allows any two of the units to be run together. The necessary gate and check valves are provided so that any of these units can be readily thrown in or out of service.

Each unit has a capacity of from 6500 to 10,500 gallons per minute against a head varying approximately from 70 to 100 feet, depending upon the level of the river which has a maximum variation between high and low water of 27 feet. The conditions to be met during the larger part of the year require each pump to discharge about 10,500 gallons per minute against a total head of 80 feet, representing an output of 212 water horsepower, which at 70 per cent. efficiency for the pump would require about 303 brake horsepower at the pump shaft. Under these conditions the head on the water turbines is approximately 50 feet which at 75 per cent. efficiency for the turbine gives 99 brake horsepower output each. Consequently there is required from the engine 204 brake horsepower, or approximately 217 indicated horsepower. The steam consumption of the engines under these conditions being 16 pounds per indicated horsepower, the duty developed per 1000 pounds of steam would be about 121 million foot-pounds. Some rough tests which were made at a capacity of about 7800 gallons per minute against about 100 feet head, conditions which are not the most efficient for either pumps or return turbines, showed nearly 110 million duty, so that it seems reasonable to believe that under the more favorable conditions above the estimated duty is actually obtained.

The normal speed of the units is 260 revolutions per minute but the engines are provided with a speed-changing device which allows of a range from 200 to 275 revolutions per minute. This is accomplished by an electric control attached to the engine governors. The engines operate ordinarily with about 130

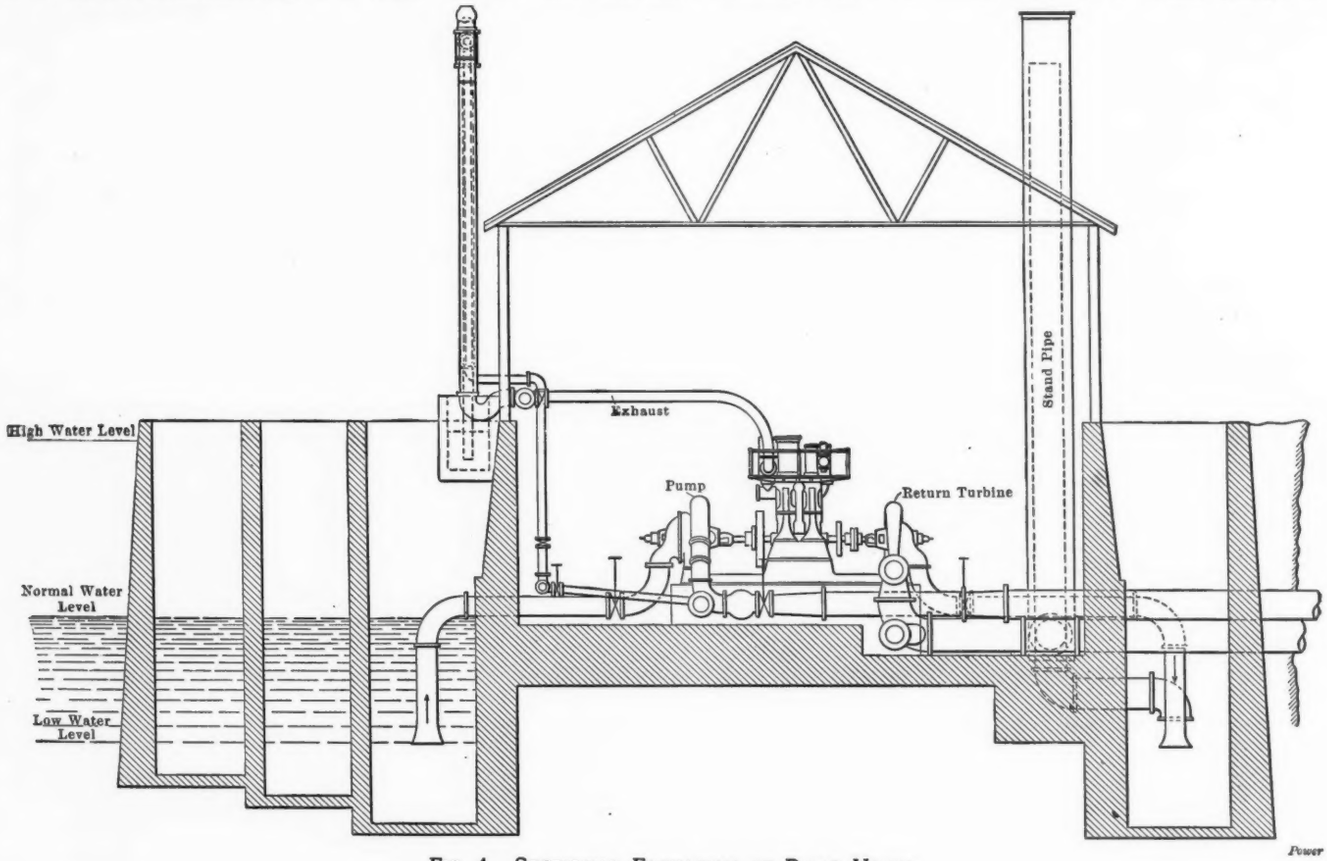


FIG. 4. SECTIONAL ELEVATION OF PUMP HOUSE

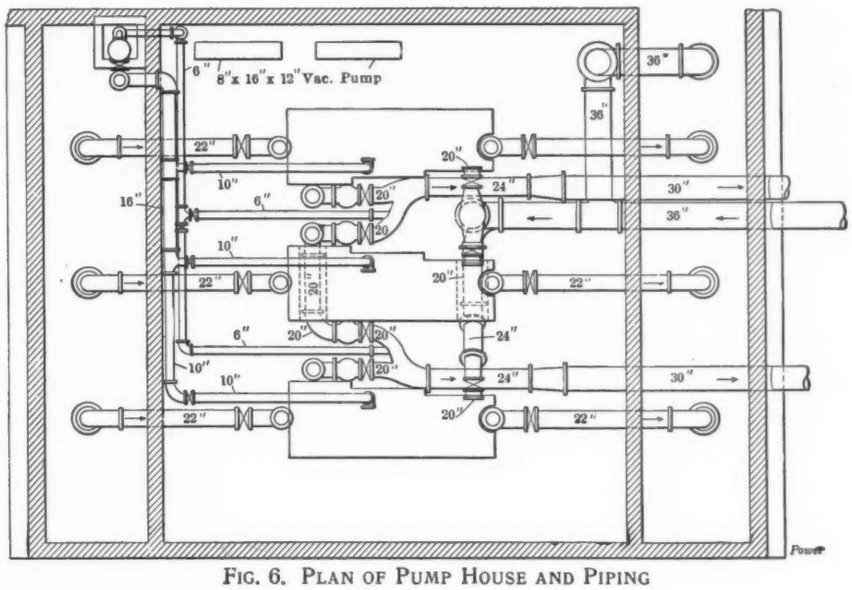


FIG. 6. PLAN OF PUMP HOUSE AND PIPING

to 150 pounds steam pressure and exhaust into one 16-inch barometric condenser, which maintains a vacuum of 26 inches with circulating water taken from the main pumps. This condenser is located as shown in Fig. 4 with its hotwell and overflow above the maximum high-water mark of the river. Vacuum is maintained by a duplicate set of 8 and 16 by 12-inch rotative dry-vacuum pumps installed in the pump house.

The return-water turbines, Fig. 1, are provided with movable gates operated by hand so that they can be set to suit the quantity of water being pumped at any time, or to suit the water level in the river, the maximum gate opening being sufficient to handle 10,500 gallons per minute at the highest river level. Under any condition to obtain the maximum power from the turbines it is only necessary to close the gates until the water level in the standpipe reaches the edge of the overflow without, however, allowing any water to go over.. A gage on the turbine readily shows when this point has been reached and as the changes of conditions are gradual and infrequent, hand regulation is all that is required. The speed of the unit is of course not appreciably affected by the turbine as the engine governor controls that entirely.

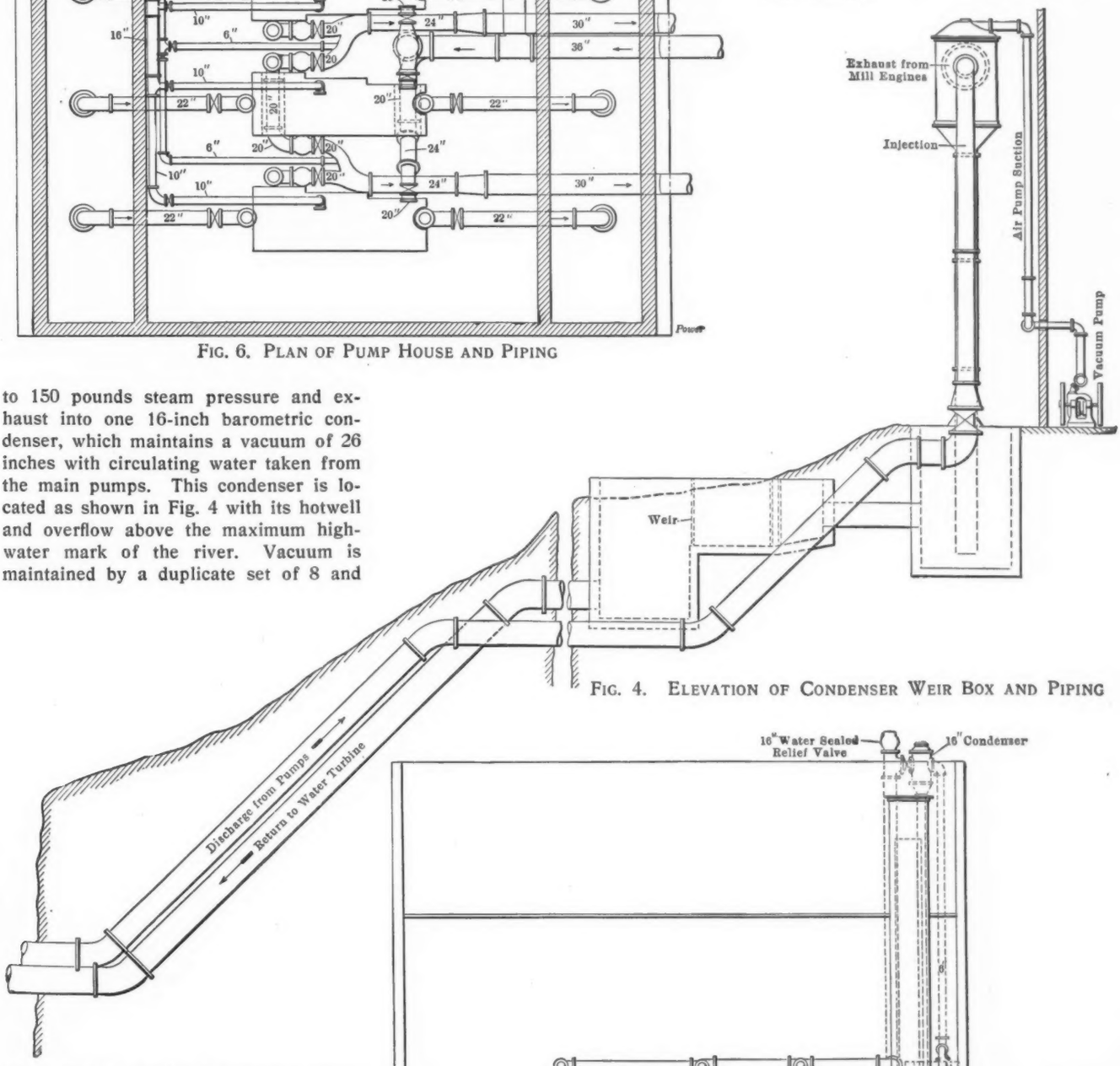


FIG. 4. ELEVATION OF CONDENSER WEIR BOX AND PIPING

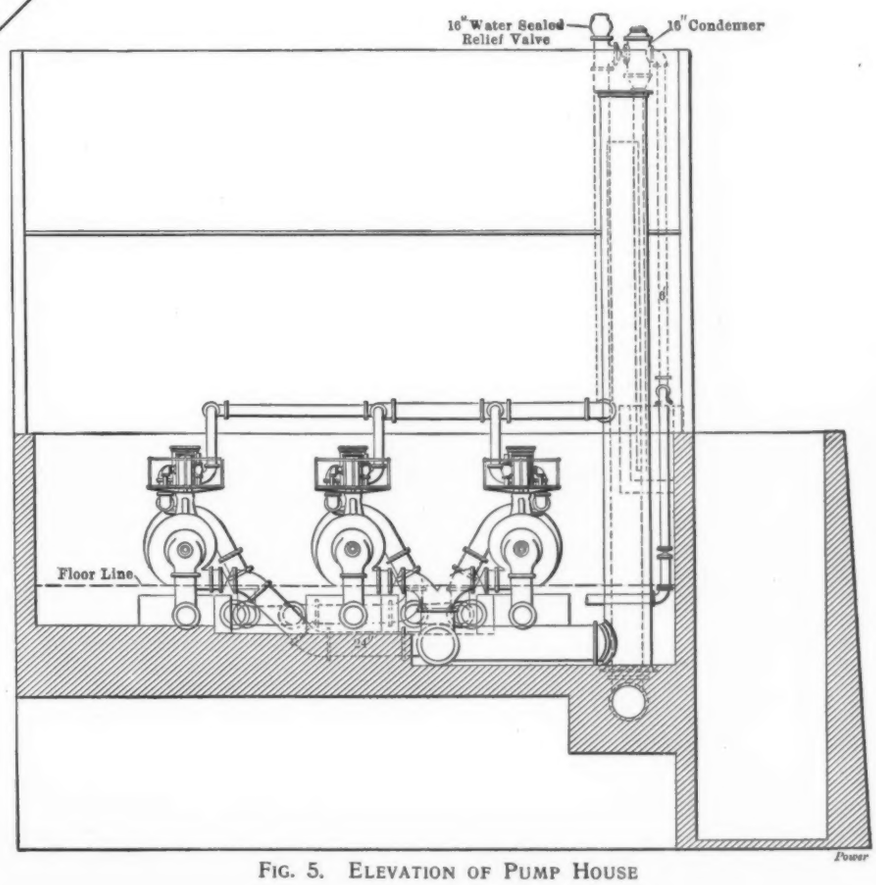


FIG. 5. ELEVATION OF PUMP HOUSE

Setting Corliss Valve Gears

Simple Methods of Setting Valve Gear on Single and Double Eccentric Corliss Engines. Each Step Taken up in Detail

BY G. J. REYNOLDS

Centralizing and Equalizing Movement of Wristplate and Rocker Arm

Suppose the valves are to be set for the first time and there are no marks on the wristplate hub to mark its central position. By the use of a plumb line, as shown in Fig. 1, set the wristplate and rocker arm in a vertical position. In this position adjust the length of the reach rod so that the hook will drop on the wristplate pin. Next mark the position of the wristplate as at *A* and *B*, Fig. 2. When these two marks are exactly opposite each other, the wristplate is at its center of motion. With the eccentric rod connected and the reach rod hooked in place, turn the eccentric on the shaft to its crank-end dead center and make a very light mark on the bracket at *C*, Fig. 3, in line with mark *A* on the wristplate. Then turn the eccentric to the opposite dead center and in like manner make the mark *D*, Fig. 4, opposite *A*. With a pair of dividers measure the distances from *B* to *C* and *B* to *D*. If the mark *D* is found to be nearer to *B* than *C*, the eccentric rod should be lengthened slightly. Another trial should be made by placing the eccentric on its crank-end dead center as before and repeating the same measurements. If it is then found that the marks *C* and *D* are equally distant from *B* the eccentric rod is adjusted to the correct length.

Equalizing Lap of Steam Valves

To equalize the lap of the steam valves, unhook the reach rod and set the wristplate in its central position. See that the hooks engage the catchblocks. Remove the back bonnets from the ends of the valve chambers so that the rear ends of the valves are exposed. On the face of each valve port a mark will usually be found showing the location of the port opening and upon the ends of the valves marks will also be found locating the opening edges of the valves. If these marks do not appear, they may be accurately located by removing the valve and laying off these lines by the aid of a machinist's square and scriber. In Fig. 5 mark *P* represents the opening edge of the valve, and *R* the edge of the steam port. When opening, this valve moves in clockwise rotation, as indicated by the arrow. When marks *P* and *R* are exactly together, the edge of the steam port and the working edge of the valve are line and line. From a table giving the lap of valves for different-sized cylinders, lay off *R S* equal to this required lap and adjust the rods from the valve-stem levers to the wristplate so that *P* and *S* are exactly opposite each other.

Adjusting Dashpot Rods for Proper Clearance between Hooks and Catchblocks

When the wristplate is in either extreme position the dashpot rods should be of such a length that the hook will engage the catchblock on the valve that is to admit steam. Turn the wristplate so that marks *A* and *C*, Fig. 3, coincide and adjust the length of the dashpot rod so that there is a very slight clearance at *N*, Fig. 6. Next turn the wristplate so that lines *A* and *D*, Fig. 4, coincide, and make similar adjustments on the other rod. This completes the adjustments on dashpot rods.

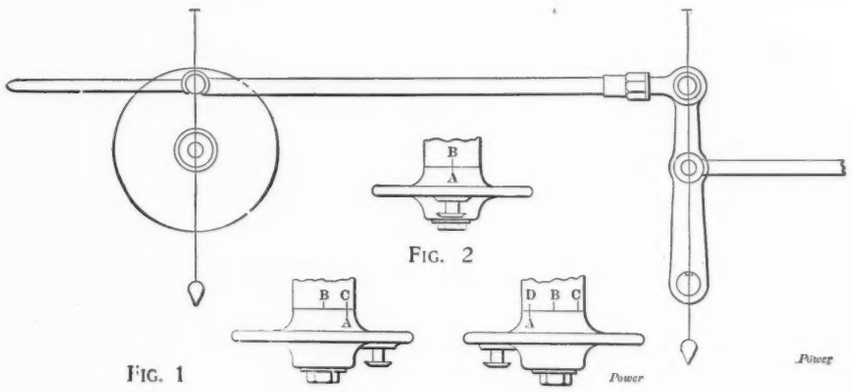


Fig. 1 Fig. 2 Fig. 3 Fig. 4

Setting Eccentric and Adjusting Lead on Steam Valves

If the head-end valve is to be set first, place the main crank on the dead center nearest the cylinder and hook the reach rod onto the wristplate stud. It will be assumed that the engine is to run over. To open the valve at the head end of the cylinder, the eccentric crank moves in the same direction as the piston; hence the eccentric is turned around on the shaft 90 degrees plus the angle of advance until the valve at the head of the cylinder has opened the port to the amount of the lead. The position of the eccentric with reference to the main crank is the same as that of a direct slide valve with direct rocker arm. The Corliss valve may be considered as direct if the reach rod in moving toward the crank end of the cylinder opens the head-end valve. If the rocker arm had been indirect the eccentric would have been set 90 degrees minus the angle of advance behind the main crank. Next tighten the set screws on the eccentric; have the crank turned to the opposite dead center and examine the lead. If it is not the same, adjust the steam-valve rod to the wristplate and see if the clearance at *N* needs to be readjusted.

Setting Exhaust Valves

When the proper amount of compression is not known, about 6 per cent. of the length of the stroke will do for trial. If the stroke of the engine is 36 inches, the exhaust-valve rods must be so adjusted as to close the exhaust valves when the piston is 2 inches from the end of its stroke. Now turn the engine in the direction it is to run until the crosshead is 2 inches from the end of its travel, as shown by marks which should be made on the guides. By adjusting the length of the exhaust-valve rod to the wristplate, the valve on the end toward which the piston is moving should be moved until the working edge of the valve comes exactly in line with the closing edge of the port. This marks the point of exhaust closure for that end. Turn the engine around in the same direction until the crosshead is 2 inches from the

other end of its travel, when the opposite exhaust valve may be set in exactly the same manner. When the wristplate is setting central, the opening edges of the exhaust valves should have a slight lap to prevent a tendency of the steam to blow through when the wristplate is in this position.

Adjusting the Governor

Most of the modern Corliss governors are fitted with a stop device or removable collar for the purpose of preventing the governor from reaching its lowest position, as when this position is reached the valves should not hook on. The stop device should be shifted so that the governor sleeve may fall to its lowest position. The wristplate should be rocked to its extreme throw each way and the safety cams adjusted so that the valves will not pick up. Should the governor belt break or the engine be very much overloaded, the governor would drop to its lowest position on the spindle, and

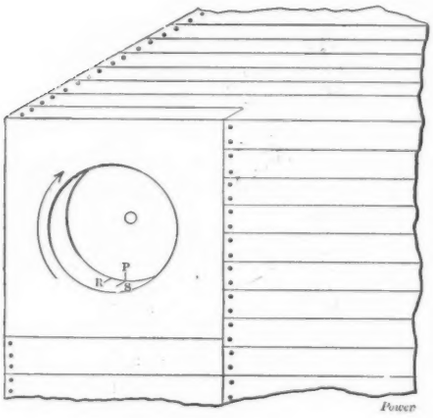


Fig. 5

as the valve cannot hook on when in this position, the admission of steam is entirely shut off from the cylinder. With the governor sleeve resting on the safety stop, the governor rods controlling the position of the knockoff cams should be so adjusted that the valves are on the point of being released by the time the wristplate reaches its extreme position. Although it is not economical to run an engine at such an overload that it will be necessary for the steam to follow the piston full stroke, this adjustment will enable the engine to deliver its maximum power.

It should now be ascertained whether or not the valves cut off steam at similar points during the forward and return strokes. With the reach rod connected to the wristplate and the governor blocked to the position of normal speed (about mid-position), turn the engine past the head-end dead center in the direction it is to run until the head-end valve is released. Measure the part or proportion of the stroke that the engine has moved from the center. Mark an equal distance from the crank center on the crosshead guides, and by turning the engine over past this center, see if the crank-end steam valve is released when the crosshead passes this line. If not, readjust the governor rod to this end to get the required point of cutoff.

The work of setting and adjusting the valves being completed, the valve-chamber bonnets may now be put on and an indicator card taken to ascertain if the setting is correct. In making final adjustments by use of the indicator, remember that the time of admission of the steam is controlled by adjusting the length of the steam-valve rods to the wristplate; time of compression and release of steam, by adjusting the length of the exhaust-valve rods to the wristplate. To equalize the cutoff, adjust the length of the governor rods controlling the position of the knockoff cams.

The Double-eccentric Engine

When single-eccentric valve setting is thoroughly understood, the double-eccentric will be found very simple. The centralization and movement of the rocker arms and wristplates will be identical with the single-eccentric engine.

To set the steam valves, place the wristplate in its central position, with the valves hooked up, and adjust the radial rods leading to the valve arms so that the valves will be open about one-fourth of the port. They should be set for such an opening that valves will have a slight lap when the dashpot rods are adjusted so that the valves will pick up when the wristplate is at its position of extreme throw each way. The reason for designing steam valves for negative lap on the double-eccentric Corliss is to increase the range of cutoff. The opening of minus lap of valves when the wristplate is central should not be so large that it will be necessary to set the eccentric less than 81 degrees ahead of the crank, or more than 99 degrees behind the crank, depending on the principle of the valve mechanism. If these limits of eccentric positions are passed, the valves will open very slowly, causing wire drawing of the steam.

With the wristplate at its extreme throw each way, the dashpot rods are now adjusted for proper clearance between the hook and the catchblock. With the main crank on the dead center, the eccentric may now be set for proper lead on the steam valves.

To set the exhaust valves the wristplate is placed in its central position and the exhaust valves adjusted so that they have the required lap as obtained from tables for different-sized engines. The engine is now turned to the point of the stroke where compression should begin. The eccentric is now shifted to the position where the exhaust valve is line and line on the end toward which the piston is moving, and there clamped to the shaft. Be sure that the eccentric is in such a position that a forward movement of the engine causes the exhaust valve to continue closing.

With the governor sleeve resting on the safety stop, hook up the steam valves alternately and adjust the governor rods leading to the knockoff cams, so that the valve which is opening will be tripped by the time the exhaust valve on the same end opens for release. If the steam valves were not released during a complete cycle of the wristplate, the valves, having no positive lap, would allow steam to blow through direct to the exhaust when the wristplate occupies a central position.

The eccentric on any Corliss engine may be properly set by making the following observations: In setting the steam eccentric, observe from the valve mechan-

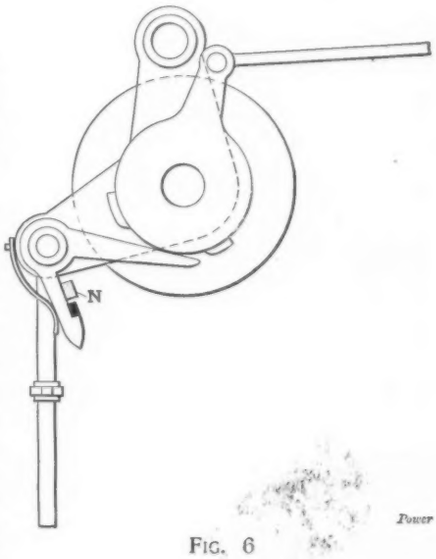


Fig. 6

ism if the eccentric crank, or full side of the eccentric, should move toward the crank end of the engine to open the head-end valve. If this is the case, the eccentric should lead the main crank. If the full side of the exhaust eccentric moves toward the crank end of the cylinder in closing the head-end valve, the eccentric must be set ahead of the crank. The exact relative position of the eccentric to the crank depends on the following suppositions:

1. If a steam valve has no lap, the eccentric must be set practically 90 degrees from the crank position, or have a very small angle of advance.

2. If the steam valves have positive lap, the eccentric must be set 90 degrees plus or minus the angle of advance, depending on the valve mechanism.

3. If the steam valves have negative lap, the eccentric must be set less than 90 degrees ahead of the crank, or more than 90 degrees behind the crank, depending on the principle of the valve mechanism.

Uncle Pegleg's Philosophy

I Learn Why Girth Seams of Cylindrical Boilers Are Single Riveted, and Incidentally Something about Figuring

After the supper table had been cleared off on the evening of the day of which I have written I sat down to wrestle with the boiler problem which Uncle Pegleg had given me.

Why is a single-riveted girth seam good enough for a cylindrical boiler, when the side seams have to be double or treble riveted?

He told me to take a plain cylinder and work it out, and I armed myself with a pencil and several sheets of foolscap and started in to do it. I assumed a 60-inch cylinder ½ inch thick and got through it by bed time, copied it off in ink and in my best writing, and was pretty well satisfied with myself.

The next day after the dinner pails had been opened up and we had got to eating, I dug down into my inside vest pocket and handed Uncle Pegleg the result of my evening's work, Fig. 1.

"Good deal of pressure to put onto a boiler head, isn't it?" he said, after he had looked it over; "282,744 pounds. Ever think what that meant? There's *over* 140 *tons* pressure on each head trying to shove them apart. That's as much as three locomotives would weigh, and all that holds them together is a ring of metal the thickness of the sheet."

I began to wonder if it could be as much as that, if I had got my figures right. He saw my uneasiness and said: "Yes, your figures seem to be all right." Taking down a book he opened it to a table which gave the circumferences and areas of circles ready made and pointed out the area for 60 inches diameter 2827.4, which was the same as mine with one less place of decimal. "But what did you multiply it by 2 for?"

"Why," I said, "there's 282,744 pounds on each head, and they are pulling against each other, a'int they? So I multiplied by 2 to get the total pull."

"Put the end of that rope in the vise. So! Now pull on it as hard as you can. So! Is that the best you can do?"

I allowed that I couldn't pull any harder.

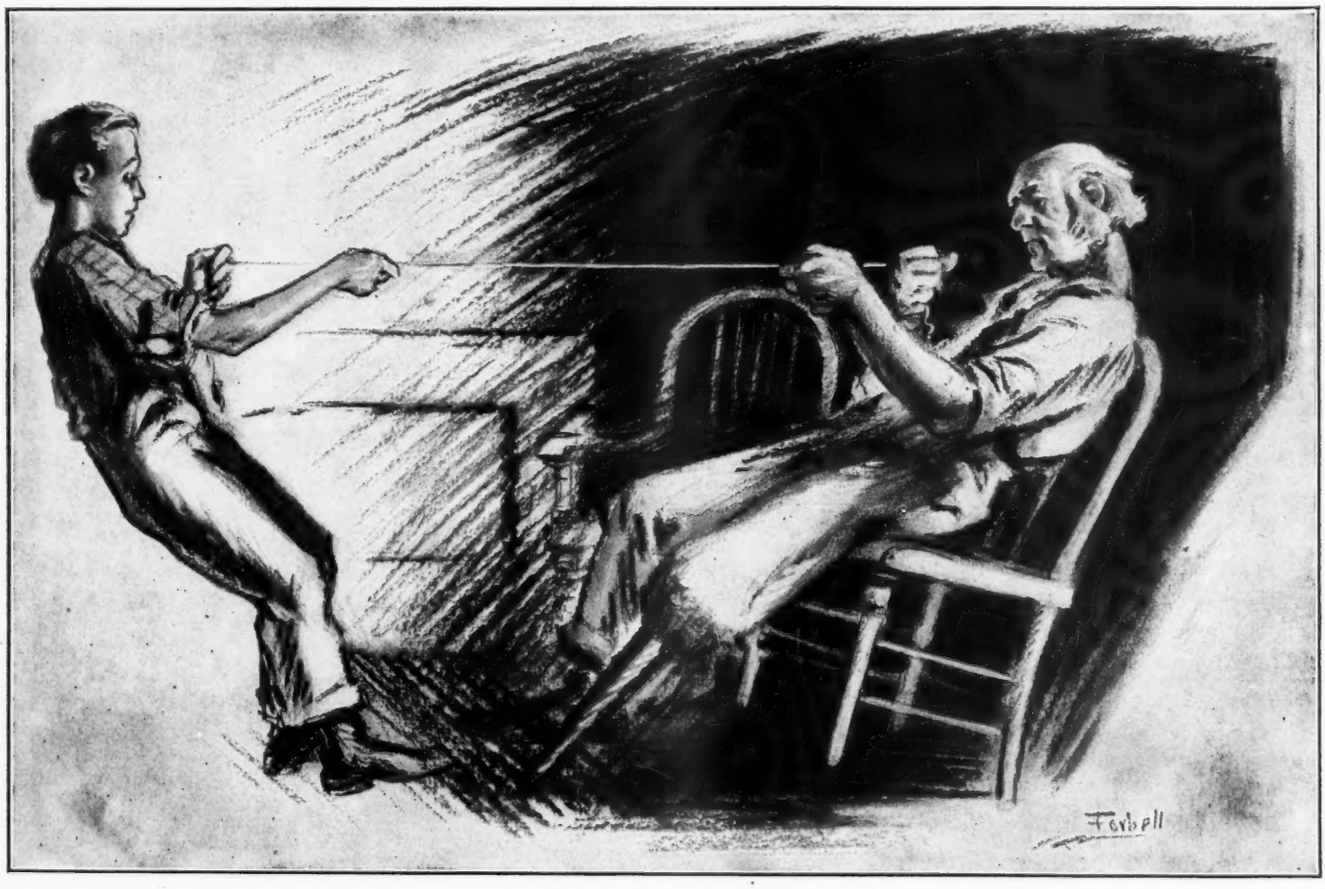
"Then there was a tension in the rope of all you can pull, eh?"

I agreed to that.

"Now, give me the end of the rope—no, the end that's in the vise."

I loosened up the vise and handed him the knotted end.

"Now, pull it," he said, and braced himself to hold against me.



"Now Pull It," He Said, "and Braced Himself to Pull Against Me"

I put all my strength into it as I had against the vise.

"Pulling as hard as you can?"

"Yep."

"Hard as you did before?"

"Yep."

"Then there must be the same tension in the rope that there was before, mustn't there?"

"Sure."

"But I was pulling as hard as you were, just exactly. If I hadn't you would have pulled me up. If I had pulled any harder than you did I would have pulled

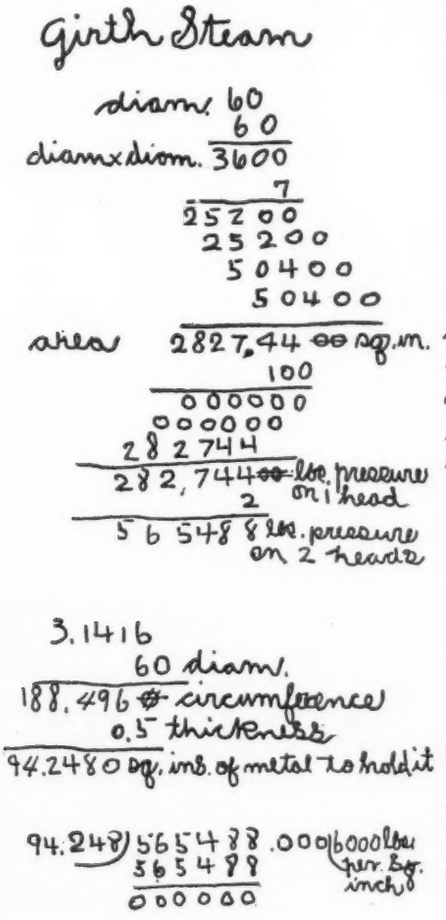


FIG. 1

you to me. Action and reaction are equal and opposite. I was just acting as the vise for you and you were acting as the vise for me, and the tension in the rope was only what *each* of us pulled. It is so with the boiler heads. This pressure of 140 tons or so acting on the front head of the boiler would shove it over into the next county if there wasn't the same pressure pushing on the rear head in the opposite direction to hold it still. The two heads are tied together by the shell and the tension in the shell is equal only to the pressure on one head, not on both. One is the vise for the other to pull against. Have you got that?"

It seemed clear enough and I told him so.

"Well, if you have, you have got one of the main principles of mechanics, that every force is opposed by an equal and opposite force. Don't let it get away from you. Now the tension on this shell is

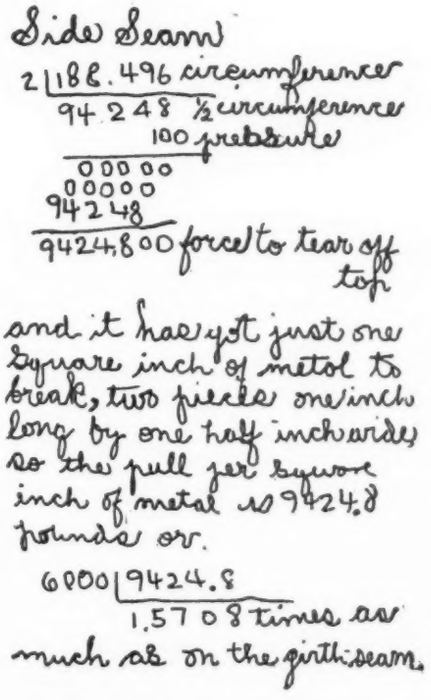
diam × diam × 0.7854 × pressure.

am I right?"

"Yes."

"How much shell is there to hold it? How many square inches of steel would be torn apart if she parted?"

"The circumference times the thickness."


Side Seam
2 | 188.496 circumference
94.248 ½ circumference
100 pressure
000000
000000
94248
9424800 force to tear off top
and it has got just one square inch of metal to break, two pieces one inch long by one half inch wide, so the pull per square inch of metal is 9424.8 pounds or.
6000 | 9424.8
1.5708 times as much as on the girth seam.

"Good! And the circumference times the thickness is

3.1416 × diam × thickness

Then the stress per square inch of metal is the total stress divided by the number of square inches of metal, eh? Or

$$\frac{\text{diam} \times \text{diam} \times 0.7854 \times \text{pressure}}{\text{diam} \times 3.1416 \times \text{thickness}}$$
$$4$$

Know how to cancel, don't you?"

"Scratch out the same things above and below the line."

"Yes, or divide or multiply a factor above and below the line by the same quantity. Now, two of the diameters cancel out and 0.7854 goes into itself once, which you don't need to put down, and into 3.1416 four times, so that the stress lengthwise on each square inch of metal through the shell is

$$\frac{\text{diam.} \times \text{pressure}}{4 \times \text{thickness}}$$

and that'll be so whatever the dimensions are, so there's no use of making a long string of figures over it.

"See how it will come out for your case. Put the figures instead of the words. Your diameter was 60, your pressure 100, and your thickness ½.

$$\frac{60 \times 100}{4 \times \frac{1}{2}} = 3000 \text{ lbs per sq. in.}$$

That's just what you would have got with all your figures if you hadn't doubled it for the two heads."

I began to see the beauty of considering quantities and their relations to each other in their natural form without having to give a definite dimension to each and figure with the dimensions.

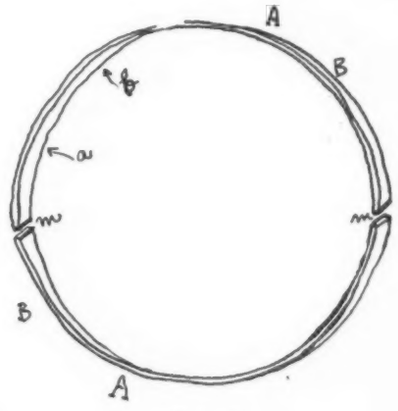


FIG. 2

"How did you get the stress in the other direction?"

"You told me I could cut the shell up into inch lengths and figure one of those, for they would all be alike."

"Did you do it?"

"Yes; I took a ring 60 inches in diameter and an inch wide, made out of ½-inch stuff and figured how much of a force 100 pounds to the square inch inside of it would get up toward busting it."

"How did you do it?"

"Well, I figured like this: You've got 100 pounds to the square inch trying to lift the top piece off from the bottom piece, Fig. 2. If the ring was straightened out it would be

60 × 3.1416 = 188.496 inches long

The top piece is

2 | 188.476
94.248 inches long

and being an inch wide has 94.248 square inches. At 100 pounds per square inch this would give a pressure of

94.248 × 100 = 94.248 lbs.

"Trying to do what?"

"To push the two halves of the ring apart."

"No, it won't. Some of that pressure is acting out this way (drawing in the arrow *a*), and trying to pull the ring apart at *A A*. Some of it is acting in this direction (arrow *b*), tending to part the ring on the line *B B*. Only a little of it acts in the direction to give a direct pull on these surfaces (*m m*)."

I scratched my head; to get all these divergent forces down to pulling in the same direction looked to me like a hopeless task. Uncle Pegleg handed me Fig. 3.

"Here is a cylinder with two pistons

in it, one perfectly flat and one like the inside of a half globe. Suppose there was 100 pounds pressure between them. You don't think the frame would move to the left because there is more area in the

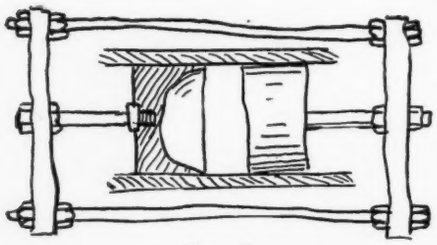


FIG. 3

hollowed-out piston than in the flat one, do you?"

It didn't seem likely.

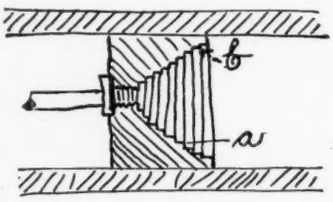


FIG. 4

"No; of course it wouldn't, because the only area that the pressure can act on is the 'projected' area. The area that would make a shadow if the lines of your force were lines of light. Suppose your piston was shaped like this (Fig. 4). Any pressure which acted on a surface like *b* would be counteracted by an equal pressure upon an opposite similar surface, but the pressure which acted upon the flat rings *a* would be unbalanced and tend to push the piston along. This would be so if the steps were so fine that a microscope would not find them and the bowl-like surface appeared regular and polished.

"You have got the same thing with your ring. It is the pressure on the diameter, not the half circumference, that you want. The force tending to separate each inch of the shell in this direction, to lift any half of the ring away from the other half, is

diam × pressure

"The number of square inches of metal which would have to be separated (as at *m m*, Fig. 2) is

2 × thickness

because there are two of them, one on each side.

"Then the stress per square inch of metal in this direction is

$$\frac{\text{diam.} \times \text{pressure}}{2 \times \text{thickness}}$$

In the lengthwise direction we found it to be

$$\frac{\text{diam} \times \text{pressure}}{4 \times \text{thickness}}$$

"I suppose you can see that the stress in the lengthwise direction is only one-half that in the other, because ¼ is only one-half of ½, and the case where you divide by 4 will give you an answer only one-half as large as when you divide by 2. Try it on your boiler."

I wrote the figures instead of the words,

$$\frac{30 \quad 200}{60 \times 100} = 6000 \text{ lbs.}$$
$$1 \times \tfrac{1}{2}$$

and found that the stress per square inch of metal along the lengthwise seam was 6000 pounds against 3000 for the round about.

"A properly made single-riveted seam is more than one-half as good as any other seam they can make, and if it's only got to stand one-half the pull what's the use of making it any better?"

And the old man stumped over to rinse out his dinner pail, while I gathered up the sketches and my discredited work of the night before.

Turbine Testing in Germany

Turbines Made by Allgemeine Elektricitäts Gesellschaft Are Given Several Days on Fully Equipped Testing Floor

BY DR. ALFRED GRADENWITZ

Every steam turbine constructed at the turbine factory of the Allgemeine Elektricitäts Gesellschaft, Berlin, is submitted, on the testing floor (Fig. 1) to comprehensive tests, lasting a few days, the load being furnished by water brakes. The testing floor is fully equipped for testing even the largest types of turbine.

The 12,000-horsepower steam turbine represented in Fig. 2 is intended for the small cruiser "Mainz" of the German Navy, and runs at a speed of 306 revolutions per minute. The "Mainz" is the first turbine cruiser to employ only two shafts, all those previously constructed being equipped with three or even four shafts. It will be readily understood that this reduction entails a most important simplification both in service and in ship construction. In fact, the design of the after part of the ship thus becomes considerably more substantial, and as each of the shafts is driven by a self-contained turbine, they are entirely independent of one another.

The turbines of the "Mainz' 'have been designed, because of their considerable length, with two housings and intermediary thrust bearings. Fig. 2 shows, to the right, the high-pressure section of the forward turbines with 12 Curtis stages, to the left, the low-pressure section of the forward turbine, and, at the left-hand extremity, the reversing turbine. As in the Allgemeine Elektricitäts Gesellschaft land-turbines, the high-pressure section comprises wheels, several rims of which are traversed successively by the same steam jet, Curtis, whereas, in the low-pressure section, there is a difference in pressure between each two rims, entailing an ever-renewed conversion of pressure into velocity. In the high-pressure section, there are formed entirely inclosed compartments, the same as in land-turbines, by intermediary bottoms, fitted into the housing, whereas, in the low-pressure section of the ship turbine, the several stages are packed at the circumference of the rim. The excess pressure between the initial steam pressure and the vacuum accordingly acts against the surface of the drums thus formed, compensating the thrust exerted by the propeller in driving the ship, and relieving the thrust bearing. The reversing turbine is designed in the same manner, the drum beginning at the pressure which, in accordance with the actual drum diameter, suffices to overcome the propeller thrust at full speed. A novel feature is the use of an air-pump driven by turbine, which affords considerable advantages in marine service.

The boiler house supplying the testing floor contains five Babcock-Wilcox water-tube boilers, of 250 square meters (2700 square feet prox.) heating area, each with superheaters and economizers. Each of these boilers generates 40 kilograms (88 pounds) of steam at 15 atmospheres, at 350 degrees Centigrade per square meter heating surface per hour though the steam temperature can be raised by superheating to 400 degrees Centigrade. Another similar unit is to be added.

This steam is also used to operate the

power house of the turbine factory, which, it seems, is the first in which all operations, even accessory duties, are done by steam turbines.

Each of the two main turbines of this power house (designed on the Allgemeine Elektricitäts Gesellschaft Curtis system) runs at a speed of 3000 revolutions per minute, being directly coupled to a 2000-kilowatt three-phase dynamo. Immediately below the main turbines are fitted below the floor, the two surface condensers, in front of each of which is arranged a pump set consisting of a steam turbine operating a cooling-water pump, and a combined centrifugal air pump and condensed-steam pump. Immediately in front of the condenser sets there are installed likewise below the floor, two boiler-feeding pumps, coupled directly to the steam turbines, and behind these, two cooling-water pumps, operated in the same manner, for the testing floor. Above the floor, there is arranged, behind the main turbines, the continuous current turbo-dynamo, of 150 kilowatts.

On account of the high speed of rotation artificial cooling is indispensable as, owing to the small armature diameter, the surfaces to be cooled are extremely small as compared with the electrical losses. This is done by air supplied by substantial fans mounted on the inductor shaft. Filters prevent any obstruction of the rotor and stator windings by the contact of the air. These filters, which consist of previous tissues, are cleaned by means of compressed air.

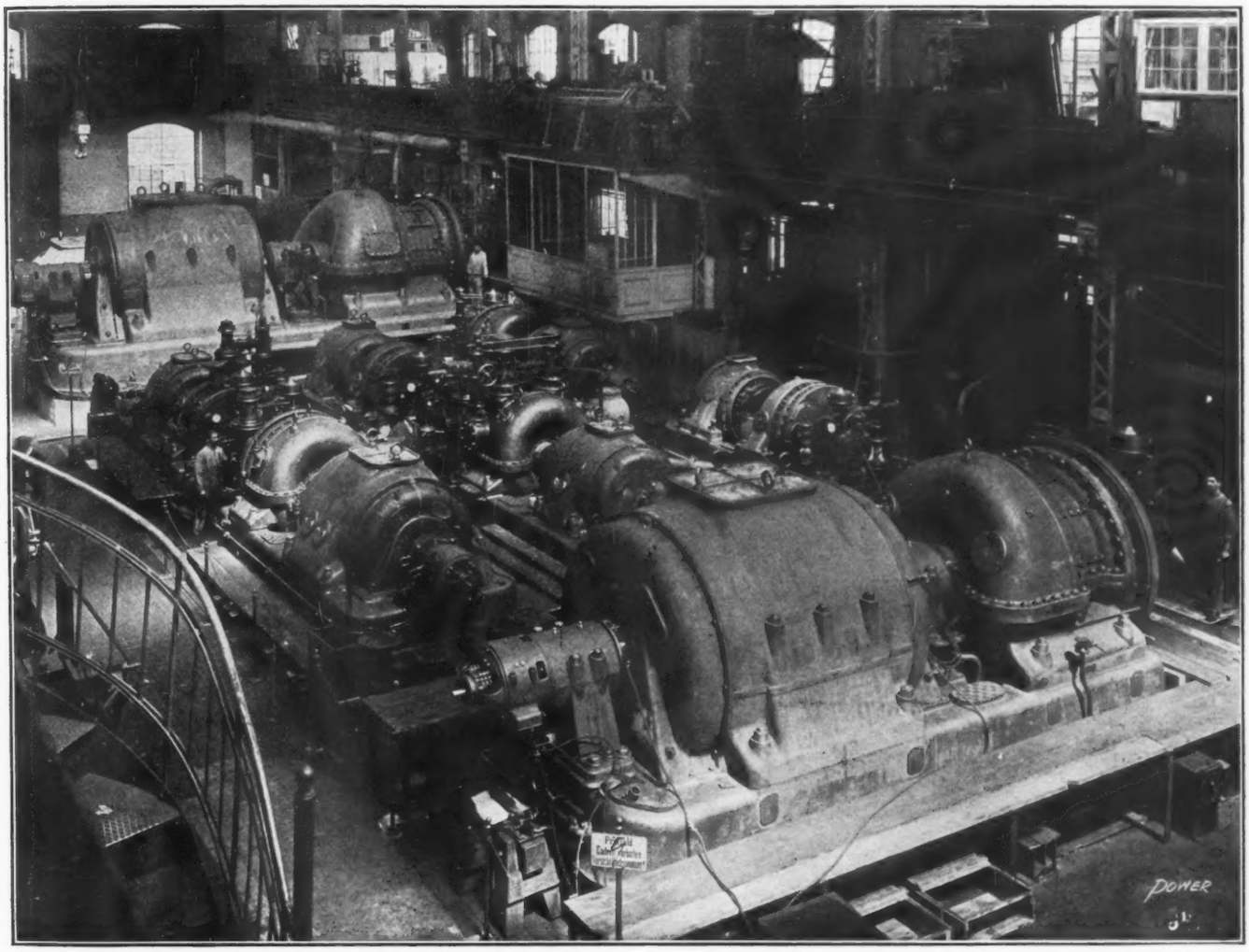


FIG. 1. TESTING FLOOR AT ALLGEMEINE ELEKTRICITATS GESELLSCHAFT, BERLIN

Strength of Steel at Different Temperatures

Reports of complete tests made at the Watertown arsenal to determine the strength of steel at different temperatures show that a certain bar had a tensile strength of 75 tons per square inch at 400 degrees Fahrenheit, but when this was raised to 1600 degrees Fahrenheit, the tensile strength was reduced to about 10 tons. The maximum strength of another bar was about 35 tons at 400 degrees Fahrenheit, but it fell to six tons at 1600 degrees Fahrenheit. Other tests show similar results, and from a due consideration of them we find that many samples of iron and steel attain their maximum strength at from 400 to 650 degrees Fahrenheit, and when this is exceeded the strength falls rapidly. Application of these results to daily practice in the boiler room shows that when our boilers have the proper amount of water in them, the material of which they are made holds to its best advantage at the highest pressure of saturated steam, but when the water level falls until some of the upper parts are exposed on one side to the direct action of heat from the furnace, with steam on the other side, the temperature attained greatly decreases the holding power of the plates, hence failures resulting from this cause, which destroy life and damage property, are not mysterious, but are the natural result of well known conditions. While it is possible for a boiler to be exposed once or twice to such dangerous conditions without causing an explosion, still the repetition of them will surely cause trouble sooner or later.

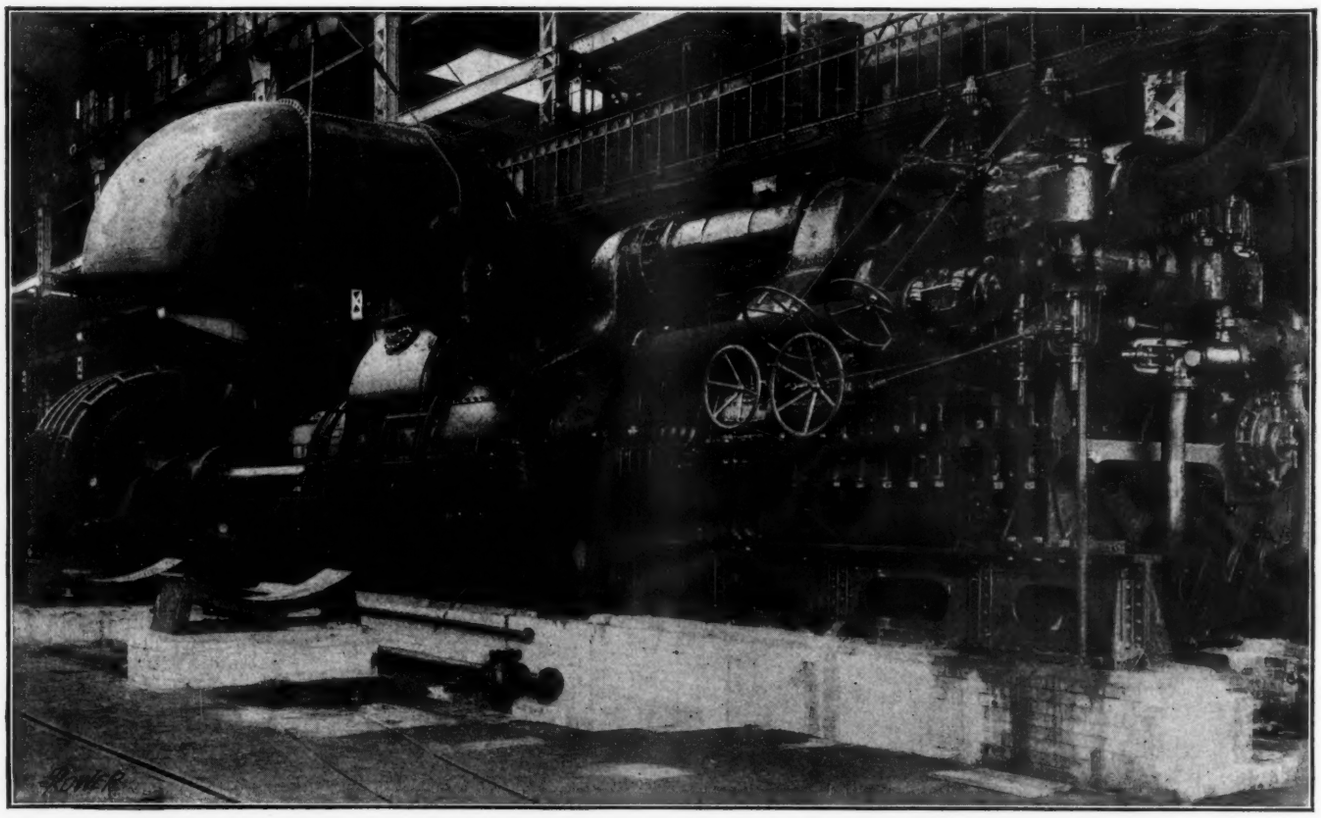


FIG. 2. 12,000-HORSEPOWER TURBINE OF THE GERMAN CRUISER "MAINZ" ON THE TESTING FLOOR

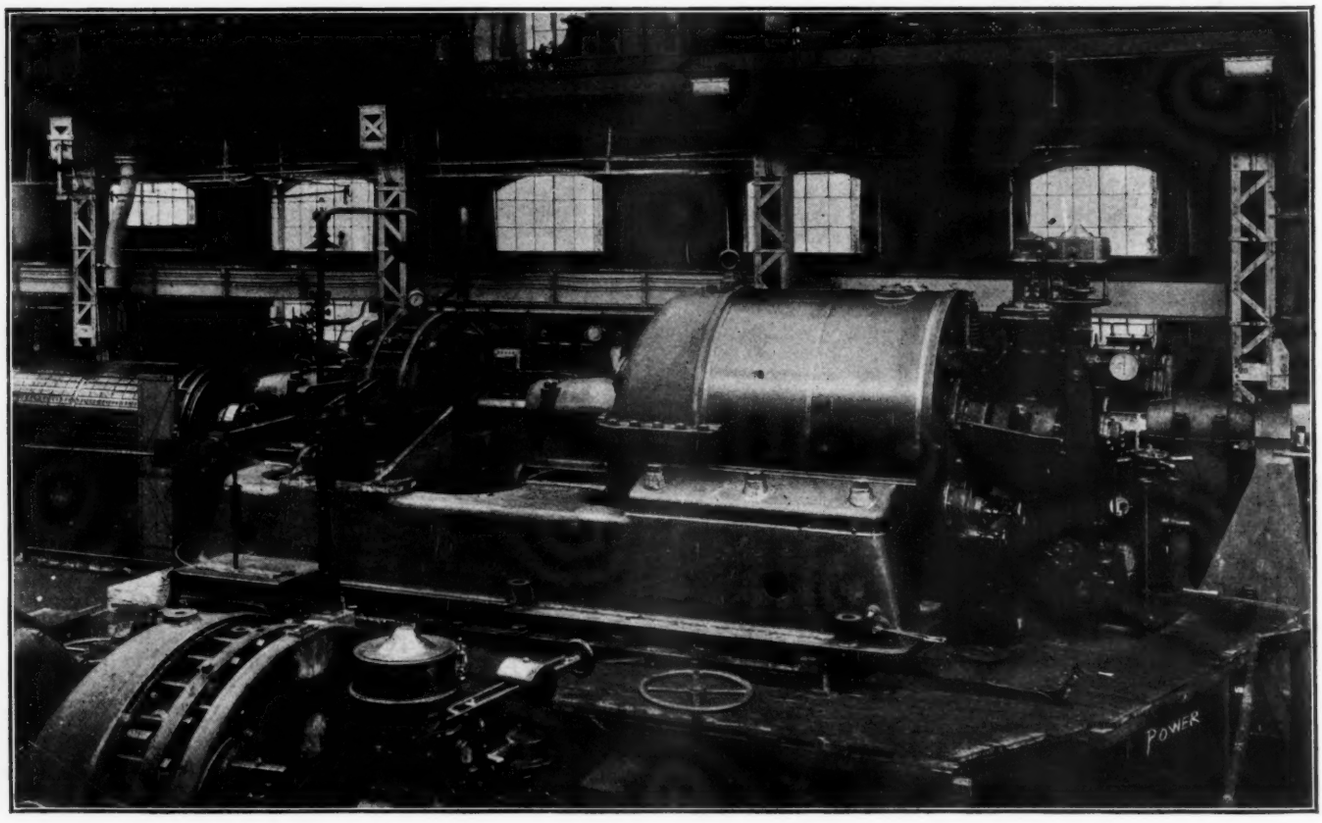


FIG. 3. TESTING A TURBINE DESIGNED TO RUN A WATER-WORKS PUMP

The Centrifugal Wet Vacuum Pump

Advantages of the Direct Driven Centrifugal as Compared with Reciprocating Type for Circulating and Hotwell Service

BY H. R. CALLAWAY

The application of centrifugal pumps to surface condensers to remove the condensed steam is of only recent date. The factor which has had the greatest influence in bringing this about is the advent of the small steam turbine. It is, of course, possible to use other prime movers for this work, but none affords as satisfactory a drive for hotwell pumps as the turbine, and it is within the last few years that the latter has been on the market. This type of installation has, however, made remarkable strides since its inception and is now at least on equal terms with the reciprocating pump, especially in large plants where space for auxiliaries is at a premium.

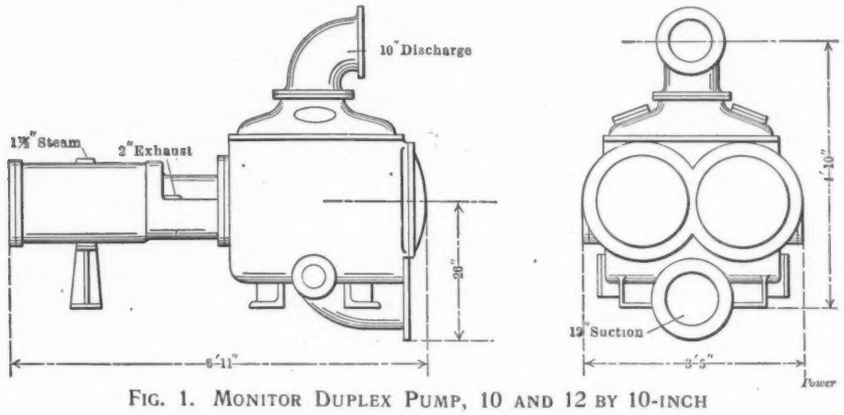


FIG. 1. MONITOR DUPLEX PUMP, 10 AND 12 BY 10-INCH

This space consideration often compels a plant designer to squeeze his hotwell pumps into a very contracted space, and under such conditions the direct-driven centrifugal is at its best. The sketches, Figs. 3 and 4, show the space taken up by two good examples and may be contrasted with that required for the plunger pumps, shown in Figs. 1 and 2. These units are all installed with 8000-kilowatt turbines, and are fair samples of such outfits for generating units. The outfit may be slid into any available place near the condenser—preferably under it—provided only that a head of at least 3 or 4 feet is allowed on the suction side of the pump. When it is possible to place the pump directly beneath the condenser and slightly to one side of or even directly under the hotwell itself, it is, of course, advantageous to do so, as this arrangement gives compactness and facilitates float-valve regulation of the pump.

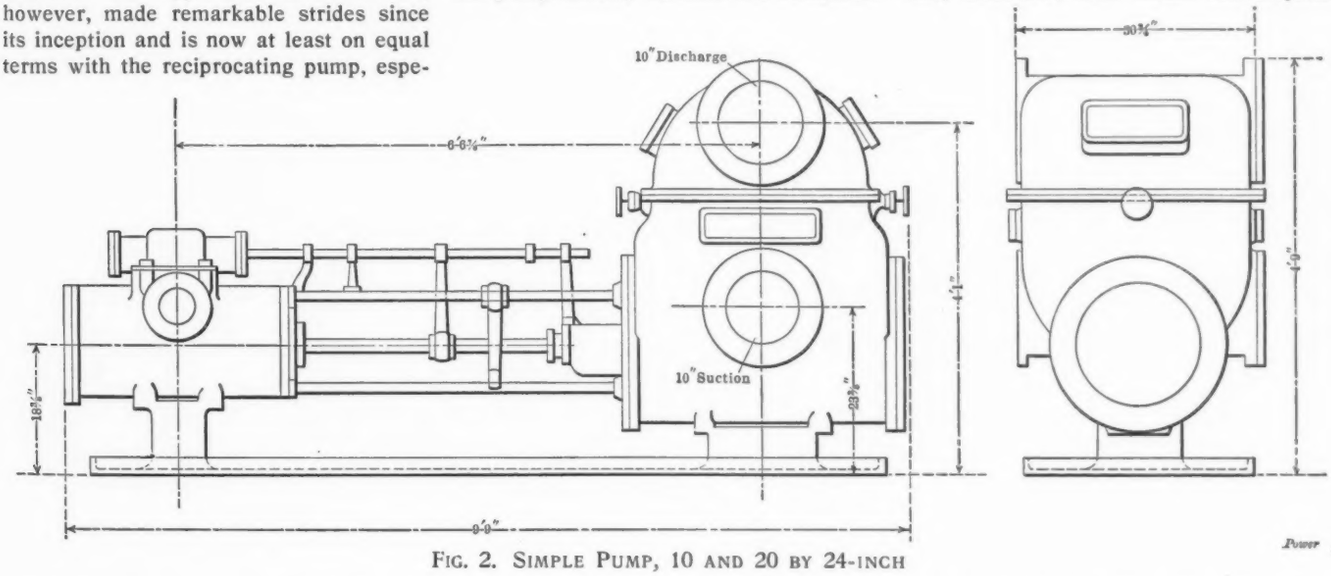


FIG. 2. SIMPLE PUMP, 10 AND 20 BY 24-INCH

Centrifugal pumps for this service must be small and consequently high-speed to handle the condensate. A 3-inch pump, for example, running at 1500 revolutions per minute will handle 300 gallons per minute against 55 feet total head, which corresponds to the steam used in an 8000-kilowatt turbine. This high speed restricts the choice of motive power to a small field. In a steam-driven station the real choice lies between electricity and steam. If the pump could run at a constant speed, a motor would be preferable in many cases, but the wet-vacuum pump's speed must vary to meet the changes in load on the main unit and in addition is subject to wide and almost instantaneous fluctuations occasioned by the operation of the float at constant load. This could be and is in some cases accomplished by means of a rheostat connected to the float arm and regulating the motor, but such an outfit is cumbersome and subject to wear and tear far exceeding that of a turbine regulator. Most of the centrifugal hotwell pumps in large stations are, therefore, direct connected to small high-speed turbines. The latter exhaust into the feed-water heaters and, therefore, their steam consumption (which is rather high under these conditions) is not a factor.

As has been said, the pumps should have at least 3 or 4 feet positive head on the suction side, as, owing to the vacuum and the high temperature of the condensate (usually within 5 or 10 degrees of the boiling point at the vacuum carried),

there is liability of vapor forming in the volute of the pump due to very slight heating. Then if the head on the suction side is insufficient the pump becomes air- or vapor-bound and refuses to remove any more water. In order to afford a steady discharge and prevent racing as much as possible, a discharge head of at least 25 feet is necessary. A loop of pipe with an atmospheric connection at the top of the loop will give the desired head if the heater is not sufficiently elevated. The discharge line should also be provided with a check valve set near the pump. Without a check, in case the pump becomes air- or vapor-bound, the vacuum in the condenser will suck the water back from the discharge through the pump and up into the body of the condenser. It is better practice to take the discharge from the top of the pump, horizontally, to minimize the chance of air pockets in the volute. Bottom horizontal discharges are more common, however, due to their convenience in piping.

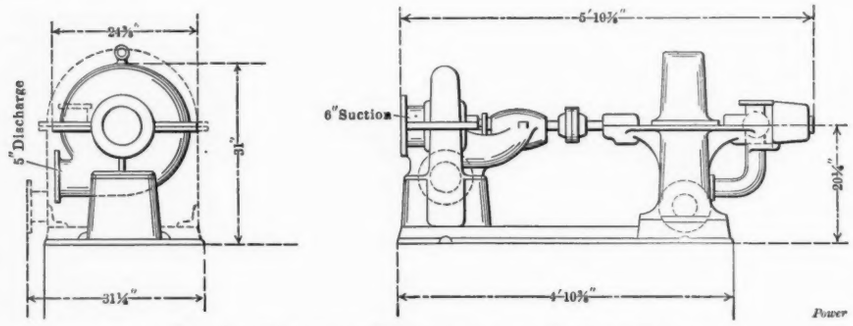


FIG. 3. FIVE-INCH TURBINE-DRIVEN VOLUTE PUMP

In operation, these units are exceedingly simple. In starting a large condensing turbine or engine, the circulating pump, dry-air pump and main unit are started in that order, and finally the hotwell pump. With a turbine-driven centrifugal outfit, merely opening the throttle of the auxiliary turbine serves to commence operations, as the turbine will start cold and the pump has absolutely no valves or other apparatus to think about. Once started, the float, check and turbine governor, Fig. 5, attend to the rest automatically. In some installations, the float valve is dispensed with, the speed being controlled entirely by the governor of the small turbine driving the pump, but the wide fluctuations in speed necessary make it difficult to adjust the revolutions so that the pump will neither empty the well entirely or get swamped. A simple ball float in the hotwell operating a balanced valve in the steam line to the turbine will keep the water level within a range of 6 inches and insure steady operation without the necessity of hand throttling. By careful adjustment of the lever arms, the level can be kept practically constant. In case of trouble the turbine governor which should be set to open at the speed corresponding to maximum load on the main unit, will prevent running away.

Once in operation, the centrifugal wet-vacuum pump requires no further attention until time to shut down, and then a few turns of the throttle-valve spindle is all that is necessary. As a labor saver this unit recommends itself to any thinking station engineer. Its extreme simplicity is one of the greatest arguments in its favor.

Because of its newness and the lack of reliable data on its operation, this type of pump has sometimes given trouble when first installed, but once these initial difficulties have been overcome, there are no further obstacles to be encountered. The effect of slight heating at high vacuum has been mentioned, and the remedy—setting the pump low enough to insure a positive head of a few feet—has already been specified. Another feature which should be carefully provided for, is absolute tightness. A very small leak when running with a high vacuum will spoil the efficiency completely. Air leaking into the pump casing will cause the pump to lose its suction and simply churn up the water without doing any work. It is always possible to discover whether the volute is full of water by opening the pet cock, from which should shoot a strong stream if the pump is working as it should. In one case it was discovered that a grease cup on the pump bearing was not tight, and every time the check valve closed the grease would be sucked into the pump and the air following it made the pump air-bound.

There is sometimes trouble with the bearings running hot, but this is a matter of adjustment and applies to any type of high-speed motor. In one installation the bearings were water-jacketed and no trouble was experienced whatsoever. If the float-valve connections to the steam valve are not properly proportioned, there may be a constant pendulum action of the water level between the two extremes of travel of the float and a corresponding speeding up and slowing down of the turbine, which gives rise to what is known as "singing." Instead of a constant quiet whirr from the machine, the note rises and falls like a siren. This may be corrected by adjusting the float levers so that a considerable rise or fall of the float causes the balanced valve to open or close a small amount only. With a little care and experimenting, the fluctuations in level may be restricted to a very slight range.

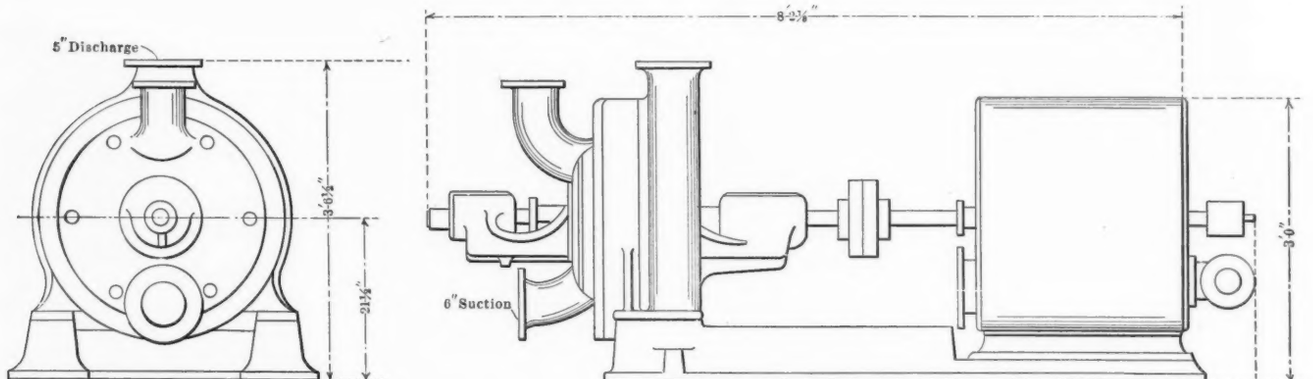


FIG. 4. FIVE-INCH TWO-STAGE CENTRIFUGAL PUMP

There have been cases where apparently everything possible had been done to insure successful operation and still the pump refused to work, and then some insignificant adjustment, done perhaps accidentally, has solved the whole trouble as mysteriously as though magic were at work. In one case that came under the writer's observation, the hotwell was separate from the condenser and a 6-inch elbow fed the former from the bottom of the condenser, as in Fig. 6. There was a float valve installed. The level in the well would remain within the zone of the float's operation, but a new level appeared in the gage glass on the condenser. In other words, the condensed water refused to more than trickle down into the well and gradually collected in the condenser instead. On breaking the vacuum, the water ran down without difficulty. The connecting elbow was removed but no obstruction could be found. The installation of the check valve had been delayed, and as there were 25 or 30 feet discharge head and the unit was badly needed, it had been decided to run without the check valve until it could be put in. After a week or so of experimenting with no success, the check valve arrived and was put in the discharge line, whereat the pump started off and ran perfectly with no recurrence of the double water level.

When no float valve is used, the pump is apt to suck the hotwell dry and become air-bound. When this happens, it is practically impossible to prime the pump, no matter how high the suction head becomes, without breaking the vacuum.

The main disadvantages in connection with centrifugal hotwell outfits are the necessarily delicate adjustments consistent with good operation, the care to be exercised to avoid leaks, and the necessity of insuring cool bearings for the

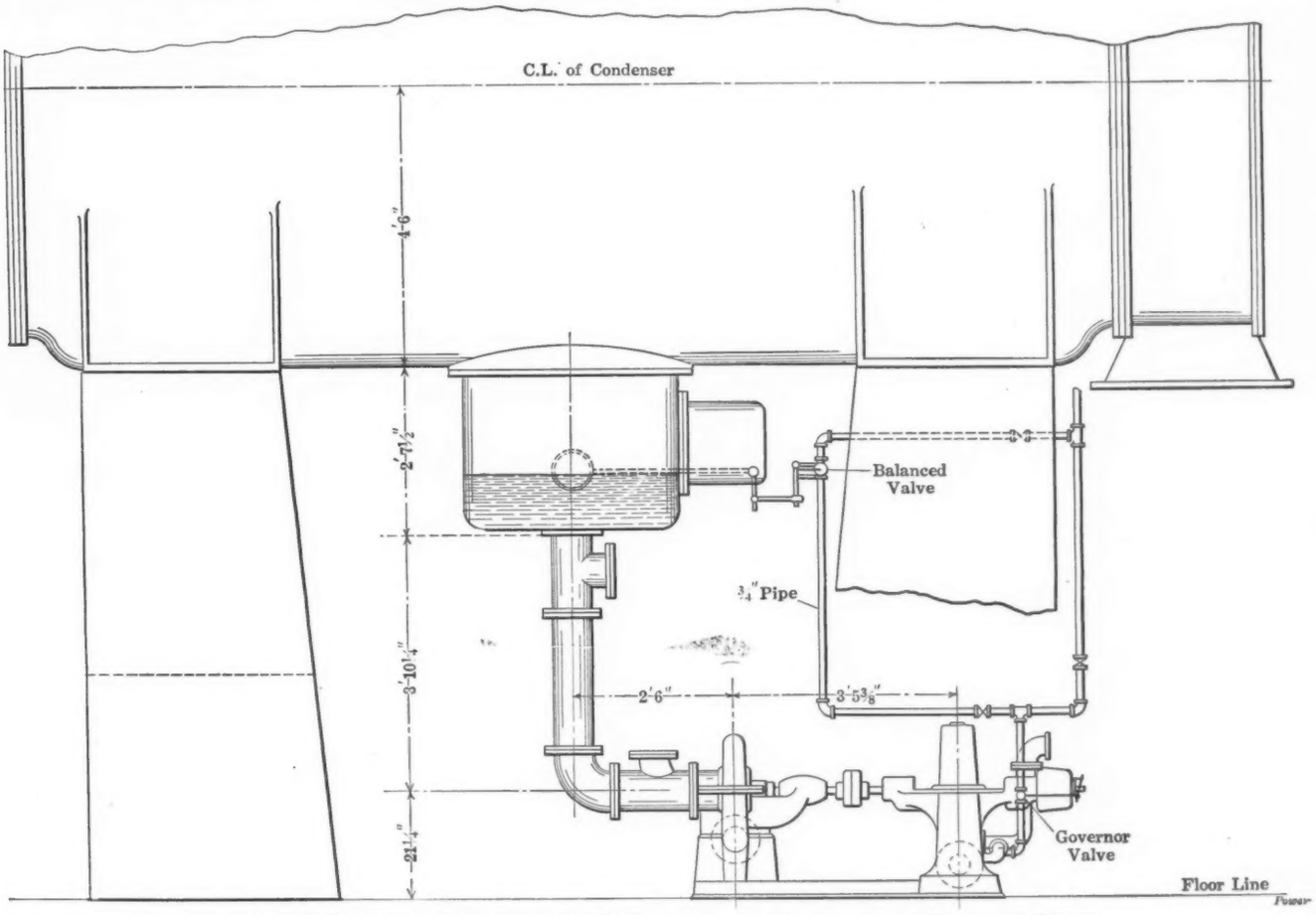


FIG. 5. STEAM CONNECTIONS FOR TURBINE-DRIVEN CENTRIFUGAL HOTWELL PUMP

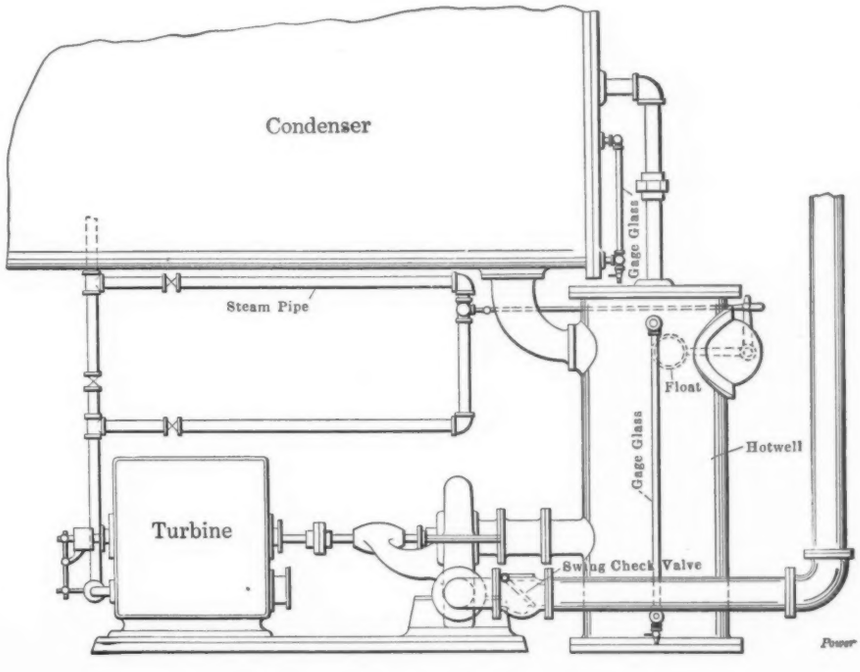


FIG. 6. HOTWELL SEPARATE FROM CONDENSER

high speeds employed. But with reasonably intelligent erection, these pumps should give very little trouble.

The advantages of this type over the reciprocating pump are many and important, and more than counterbalance whatever troubles may arise incident to their operation. Very often the space consideration is of sufficient importance to condemn the reciprocating pump without looking any further. For testing, the centrifugal pump lends itself to the use of the Venturi meter. The slow pulsations of the plunger pump make it impossible to measure the condensed water by this means. As already stated, when the turbine-driven centrifugal pump has been started, the attendance required is practically nil. The saving in labor consequently is an item in a large plant. The extreme flexibility of this outfit is very much in its favor for hotwell service, as it will respond almost at once to wide fluctuations in load and is perfectly automatic in its action. In case of a sudden peak the operating force has plenty to do without thinking about the wet-vacuum pump.

With centrifugal circulating and hotwell pumps the attention demanded by auxiliaries is materially reduced, and with the advent of a practical centrifugal dry-air pump for high vacua, will reach a minimum. It might then be possible to drive the dry-air and circulating pumps from the same turbine, with a considerable saving in space and attendance.

Mean Effective Pressures Do Not Check

Pressures Actually Obtained from Single-Cylinder Noncondensing Steam Engines Are Lower Than Usual Tabular Values

BY JAMES B. STANWOOD

It has been the experience of engine designers and engineers that the mean effective pressures obtained in practice from noncondensing simple steam engines do not correspond with those usually given by the tables that are in vogue and are published by different engine builders.

Whenever it is desirable to know the probable mean effective pressure for a given pressure in the steam pipe near the engine throttle valve for a given actual cutoff, Table 1 is presented as being more in accordance with general practice. The drop in steam pressure from the steam-pipe pressure to the point of cutoff is considered as an essential factor and is recognized in these calculations; it is assumed to be 20 per cent. of the absolute steam pressure in the steam pipe. This amount in some cases is excessive, but by comparing a large number of indicator cards taken under different conditions and from different types of engines, this is not an extreme amount. This is especially true since the advent of high-speed engines with valve gears operated by main-shaft governors, in which the closing period of the valve is not so sharply determined on the diagram as with slow-speed Corliss engines or engines fitted with gridiron valves.

The basis of figuring these mean effective pressures is shown in the accompanying diagram. The following assumptions are the basis of the calculations:

Drop of cutoff pressure, 20 per cent. of the absolute steam-pipe pressure.

Clearance space, 5 per cent.

Compression and release take place at 85 per cent. of the stroke.

Back pressure, 15 pounds absolute.

The effective area, which can be obtained, consists of the large area X, or $A\,B\,C\,D\,E$ plus the triangle T, or $F\,G\,B$, less the compression area S, the back-pressure area W up to the point of compression, the area of loss after release V, or the triangle $M\,C\,O$, and the area of the unfilled clearance space U, or $A\,F\,Y\,Z$. Expressed in a formula,

$$\textit{Effective area} = X + T - (W + S + V + U).$$

The compression area S is equivalent to

TABLE 1. OBTAINABLE MEAN EFFECTIVE PRESSURES IN NONCONDENSING STEAM ENGINES.

Steam Pipe Pressures Near Throttle. Pounds.	Actual Cutoffs of Admission Valves.									
	0.15	0.2	0.25	0.3	0.35	0.4	0.45	0.5	0.55	0.6
60	11.3	16.0	19.6	23.1	25.9	28.5	30.6	32.6	34.2	35.5
70	15.4	20.6	24.8	28.6	31.8	34.8	37.2	39.5	41.3	42.7
80	19.5	25.1	29.9	34.1	37.8	41.0	43.8	46.3	48.5	50.0
90	23.4	29.6	34.9	39.6	43.6	47.2	50.4	53.0	55.5	57.4
100	27.3	34.2	40.0	45.2	49.6	53.6	57.0	59.9	62.7	64.7
110	31.3	38.7	45.0	50.7	55.6	59.8	63.7	66.7	69.7	72.0
120	35.2	43.2	50.1	56.2	61.4	66.0	70.2	73.6	76.9	79.3
130	39.1	47.7	55.1	61.8	67.3	72.2	76.8	80.5	84.1	86.7
140	43.0	52.2	60.2	67.4	73.3	78.5	83.4	87.5	91.3	94.2
150	46.9	56.9	65.2	72.9	79.3	84.9	90.0	94.4	98.3	101.6

TABLE 2. COMPARISON OF MEAN EFFECTIVE PRESSURES OBTAINED IN PRACTICE WITH TABULAR AMOUNTS.

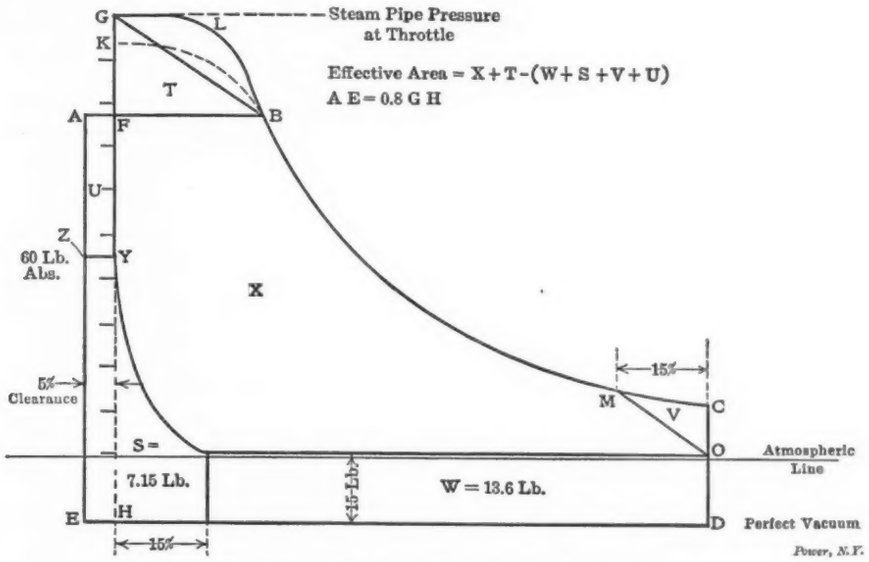
1	2	3	4	5	6	7
Type of Engine.	Size, Inches.	Steam Pipe Pressure, Lb. Gage.	Actual Cutoff.	Actual M.E.P. from Diagrams, Lb. per Sq.In.	M.E.P. as per Table 1.	M.E.P. as per Usual M'f'r's Tables.
Single valve....	8x12	83 82.4 81.9	0.203 0.312 0.378	24.5 34.8 43.1	26.2 35 40	36 48 55
Single valve....	13x12	100	0.32 0.42 0.53	45.5 56.5 67.6	47 55 61	64 76 84
Corliss.....	23x60	72.8	0.367	33.1	33	46
Corliss.....	28¼x59½	101	0.315	41.2	46	61
Corliss.....	16x42	100	0.178 0.231 0.323	29.5 38.1 48.4	30.5 38 47.4	40 49 64
Corliss.....	22x30	148.5	0.201	54.6	55.5	..
Gridiron valve.	28x60	65.1	0.222	23.8	20.8	30
Four-valve.....	19x18	100	0.185 0.285	42.3 29.3	44 32	58 42

7.15 pounds mean effective pressure, the back-pressure area *W* is equivalent to 13.6 pounds, and areas *U* and *V* are variable, as are also the large area *X* and triangle *T*.

In practice the steam line follows the curve *G L B* from initial pressure, but in this consideration it has been taken to follow the hypotenuse of triangle *T*. The difference between the two lines will offset the usual drop in initial pressure, as shown from *G* to *K*.

The line *G F* is just two-tenths as long as the line *G H*, or, as previously stated, the drop in steam pressure from initial to the point of cutoff represents a drop of 20 per cent. of the absolute steam pressure in the steam pipe.

To show the correspondence of this table of mean effective pressures with diagrams in practice, Table 2, which gives the style of engine, cutoff, steam-pipe pressure and the actual obtainable mean effective pressure for each case, is submitted. Column 6 gives the mean effective pressure as given by Table 1. In the last column are shown the mean effective pressures which are found in a well known table of a prominent engine builder. The discrepancies between actual mean effective pressures and those given by these values are excessive, and the usual tables are therefore misleading.



EFFECTIVE AREA OF INDICATOR DIAGRAM

6000 H. P. Absorption Dynamometer

Principle of Reversed Turbine Used in the Solution of New Problem in Water Brake Design. Large Power at Slow Speed

For measuring the power output of prime movers, such as steam engines and electric motors, the prony brake has for a long time been considered a satisfactory piece of experimental apparatus when handling moderate powers. This device, as is fairly well known, usually consists of bands of rope put around the rim of the flywheel, these bands being attached to a frame resting upon a balance scale, the wheel revolving inside the bands, which remain stationary. Suitable means are provided for applying tension to the ropes to make more friction on the flywheel rim and thus increase the resistance to the rotation of the flywheel and consequently the pressure upon the scale.

In this way it is clearly seen that the power of the machine to be tested is transformed into heat due to the friction of the bands and this heat is carried off by a stream of water which flows into the inside of the rim, is carried around by the centrifugal force and finally scooped out at some other point by a suitable overflow pipe. Another evidence of the friction produced is the tendency to rotate the band, and this is measured by the downward pressure upon the scale which resists this rotating tendency.

To calculate the horsepower delivered it is assumed that the force measured by the scale is acting at a lever arm equal to the distance from the center of the wheel to the point of support at the scale, and that if the band were allowed to rotate, it would make as many revolutions per minute as the wheel does. The two elements of work, that is, force (the pressure upon the scale) and the distance traveled in a given time (the circumference corresponding to the lever-arm radius, multiplied by the revolutions per minute) are then both known, and their multiplication will give the power being absorbed by the prony brake.

WATER BRAKES

Such a device as this is suitable for moderate power, but when more than about 200 horsepower is required to be measured, the heat generated becomes excessive and the difficulties of constructing the brake with sufficient area to carry off this heat make it a practical impossibility. For the purpose of testing steam turbines, manufacturers have for some time been using what is known as a water brake. This device uses practically the same principle as the prony brake, with the exception that the energy developed by the turbine is absorbed in churning up a quantity of water so that its temperature is raised, and in fact some of the water is boiled away. This device is suitable for much larger powers than the prony brake.

In a water brake the essential features are a rotor mounted upon the shaft which is delivering the power, a stator which surrounds the rotor, and a quantity of water between the two. The stator is provided with a lever arm which protrudes from one side, so that it may rest upon a scale, the weight of the stator body being supported by bearings on the revolving shafts. In the periphery of the rotor, which is of cast iron, there are blades, somewhat like those of a paddle wheel, with the exception that they are inclined forward, and furthermore, are in two diagonal sections, something like the teeth of a herringbone gear. Surrounding the rotor there is the cast-iron shell of the stator and on the inside of this there are other blades arranged somewhat like the teeth of an internal ratchet. These teeth are not very deep, but are faced in such a direction that the water which is admitted between the revolving and stationary elements, is thrown with great violence by the rotating blades against the faces of the ratchet teeth on the inside of the stator. This violent throwing of the water not only heats it up and causes some of it to boil away, but the impact of the water against the blades of the stator causes the same tendency to rotate that there is in the ordinary prony brake. This tendency shows itself in a force at the end of the lever arm bearing down upon the scale. Again, in this de-

vice, by noting the pressure on the scale and the speed of the shaft, the horsepower is easily calculated. The load upon the machine is adjusted by the amount of water admitted to the brake, and this is controlled by valves.

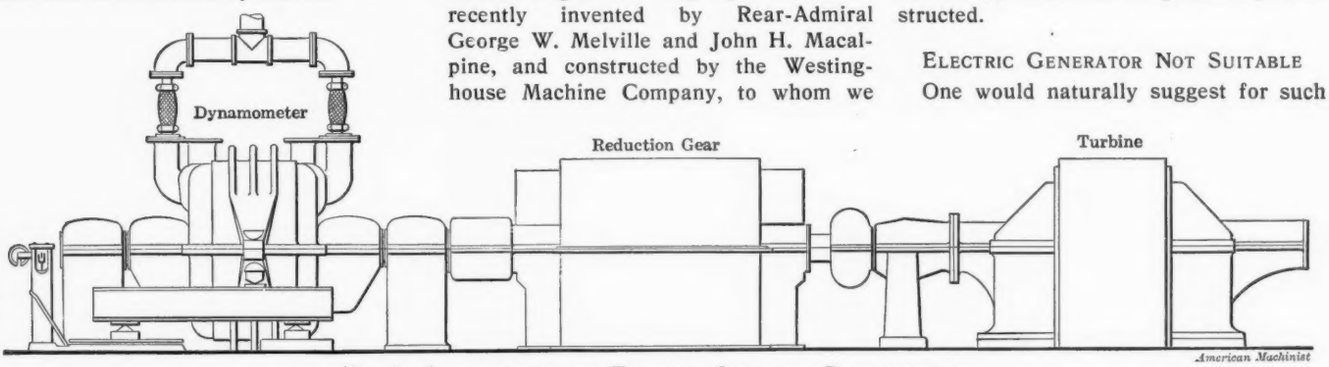


FIG. 1. ARRANGEMENT OF TURBINE, GEAR AND DYNAMOMETER

THE NEW PROBLEM

These water brakes are suitable for high speeds; that is, from 750 revolutions per minute to 4000 or more, the former figure being for large machines and the latter for small ones. Of late, however, there has been a call for a water brake involving a new problem in the fact that it would be necessary to absorb 6000 effective horsepower at the comparatively low speed of 300 revolutions per minute.

Necessity for this brake was due to the desire to measure the capacity and efficiency of the experimental spur-wheel reduction gear for high-speed turbines recently invented by Rear-Admiral George W. Melville and John H. Macalpine, and constructed by the Westinghouse Machine Company, to whom we are indebted for the accompanying illustrations. A brake for such conditions as this was entirely unprecedented. First, the development of 6000 horsepower, if transformed into heat, is practically equal to that which would be generated in a furnace burning coal continuously at the rate of 1200 pounds per hour. If such a brake could be constructed in the prony type the speed of 300 revolutions per minute would limit the diameter of the brake wheel to 6 or 7 feet, and in order to get sufficient surface to absorb the power, the face of this wheel would have to be at least 30 feet long. Such a brake, of course, could not be practically constructed.

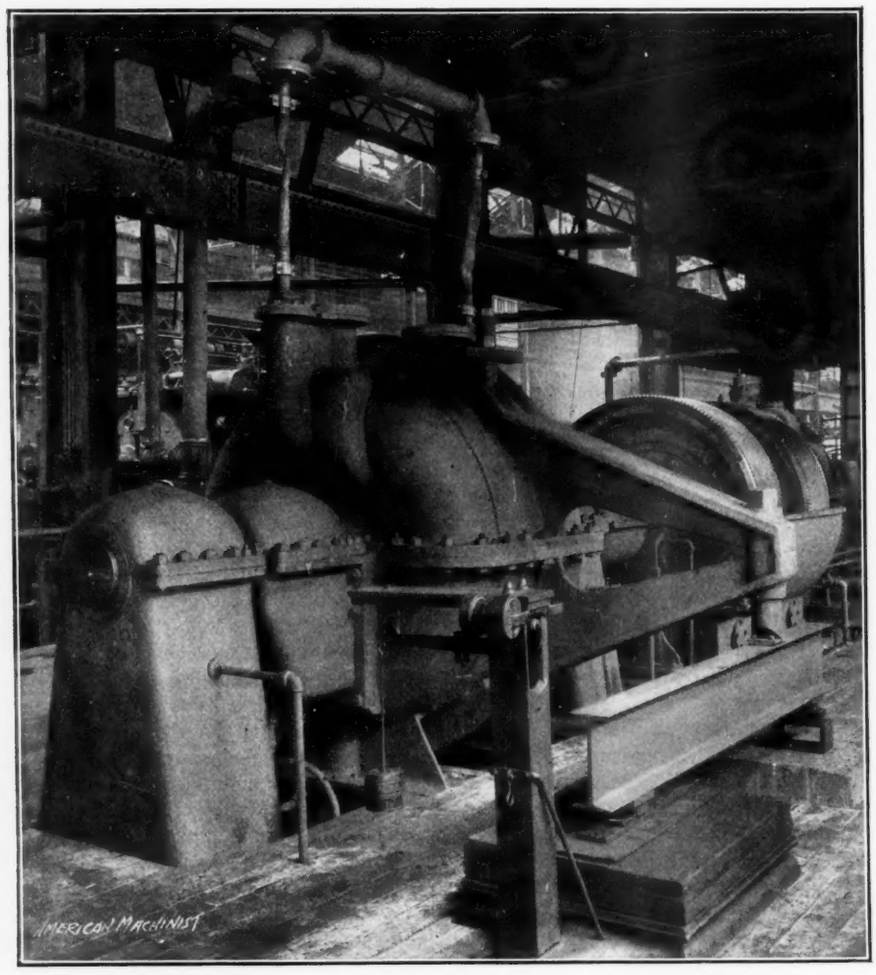


FIG. 2. DYNAMOMETER ASSEMBLED FOR THE TEST

ELECTRIC GENERATOR NOT SUITABLE

One would naturally suggest for such conditions as this, that the power be absorbed by an electric generator, the current developed being passed through water resistance, in order to vary the load as desired. The trouble with this is that the generator would be cumbersome and expensive, and many of the losses such as windage, eddy currents, etc., in the generator itself make the measurement of the power subject to certain assumptions which to a considerable extent invalidate the object of the tests, namely, to accurately measure the output and also the efficiency of the gearing.

SOLUTION: THE TURBINE BRAKE

Such an electric generator not being suitable, the idea of using the old-style water brake would, of course, suggest itself, but it was recognized that the excessive power at the extremely low speed would necessitate some change in design. This is exactly what was settled upon, and the hydraulic brake, herewith illustrated, was developed by Raymond N. Ehrhart, of the Westinghouse engineering staff, for this purpose. The principle invoved is practically the same as that of the ordinary water brake, with the exception that the principle of the steam turbine, reversed, was incorporated; the object being to get the violent churning effect at the lower speed and to obtain it in a machine of practical size.

Fig. 2 shows the general outside view of the apparatus assembled for experiment, the part in the foreground, looking not unlike a large centrifugal pump, is the stator casing of the dynamometer. At the right will be seen the lever arm extending from the shell and resting with a knife edge upon an I-beam, which in turn rests upon a fulcrum in the distance and the scale in the foreground. By this double-lever system the pressure of the rotating tendency is reduced in the ratio of 1 to 4, and so is easily and accurately measured upon standard scales. In Fig. 2, in the distance, also will be seen the two large spur gears of the Melville-

Macalpine device. These gears are mounted directly on the shaft upon which is also mounted the rotor of the dynamometer.

At the top of the stator will be seen the two rubber-hose pipes through which water is admitted to the dynamometer casing. That portion of the water which is transformed into steam by the violent churning action comes out through the top of the shell where two bank flanges are seen in the halftone. The water which does not boil away passes out through two pipes at the bottom of the casing, the openings being regulated by valves on the ends of these pipes. There are two objections to allowing the energy to be used in boiling all the water in the casing. In the first place, if it should all boil away suddenly the load would be at once removed from the dynamometer, and secondly, the violent boiling action makes it difficult to regulate the quantity of water in the casing and makes the pressure upon the scale very irregular, so that uniform results are difficult to obtain. When in operation, an attendant carefully adjusts the amount of water flowing so that the pressure upon the scale beam is just balanced at a constant point, thus showing that the load transmitted by the gears tested is also constant.

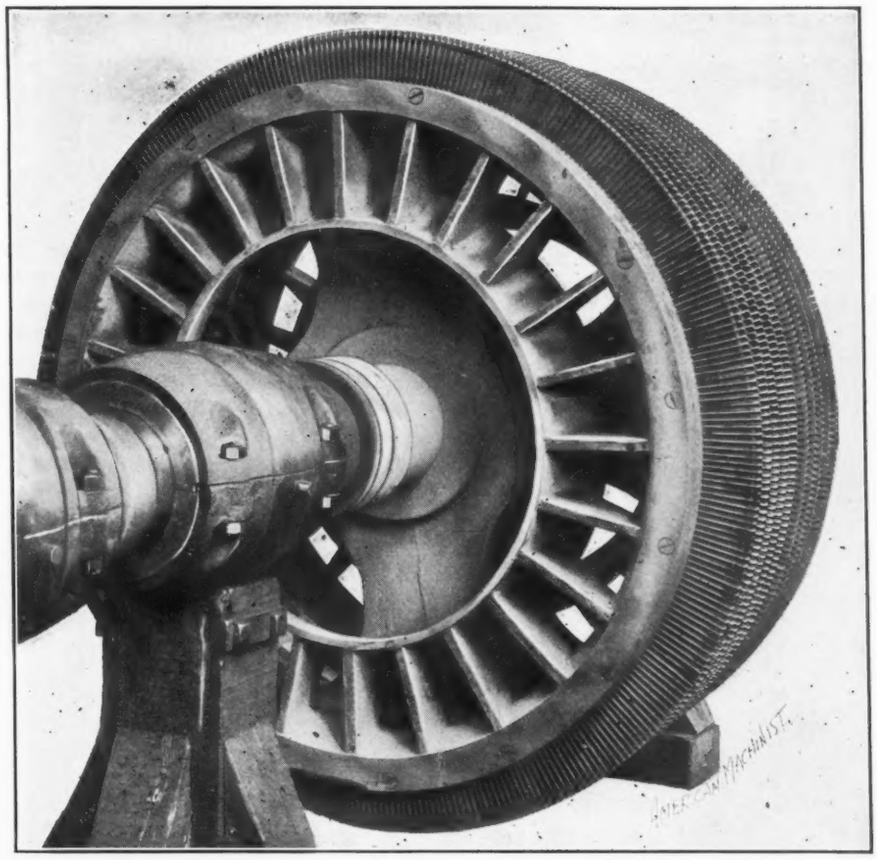


FIG. 3. SIDE VIEW OF ROTOR SHOWING WATER PASSAGES

As with the ordinary water brake, the load may be varied by adjusting the water supply. Fig. 1 shows the general arrangement of the apparatus, the steam turbine being on the right and the reduction gear in the middle with the dynamometer at the left. Fig. 3 shows a side view of the rotor, the large grid openings in the side being those shown at *F* in Fig. 5. In Fig. 3 also, are clearly seen the turbine blades in the periphery of the rotor. These blades, as in turbine design, are dovetailed into the rotor frame and laced together near the outer

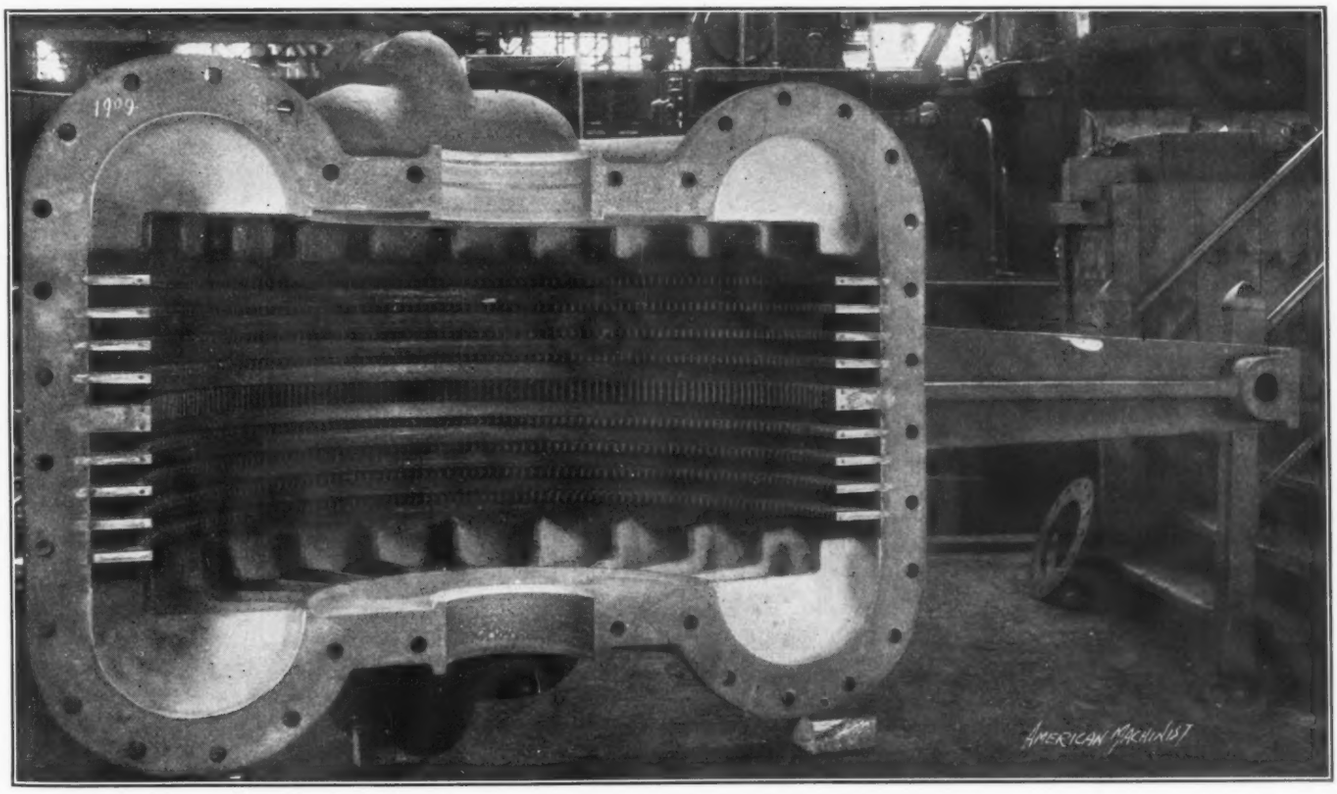


FIG. 4. INSIDE VIEW OF STATOR SHOWING FLAT STATIONARY BLADES INSIDE AND LEVER ARM AT RIGHT

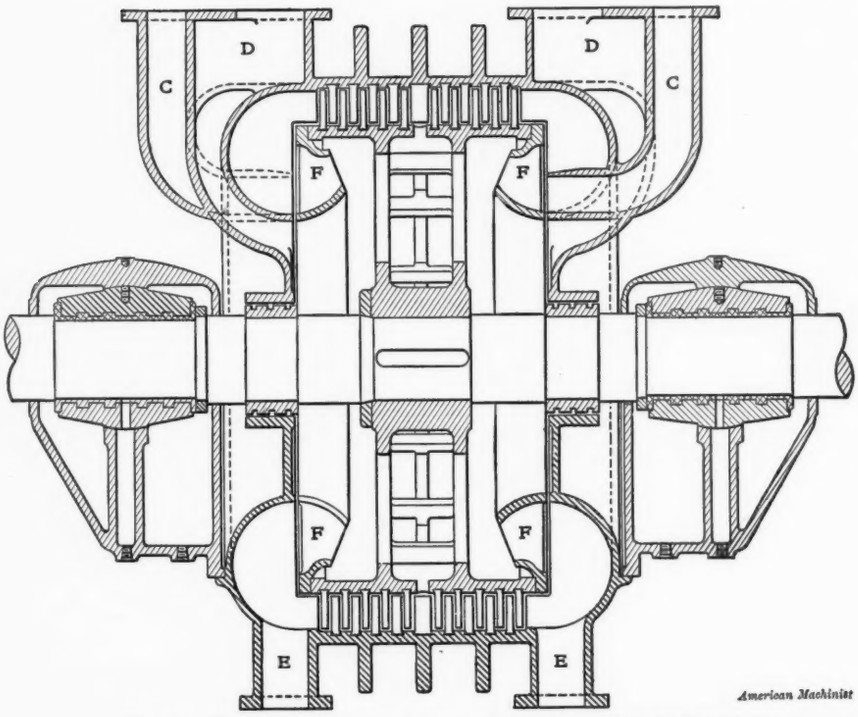


FIG. 5. CROSS-SECTION THROUGH ASSEMBLED DYNAMOMETER

ends by the wire shown in the halftone. This wire is of peculiar shape, being not unlike a comma in section. The holes punched through the blades are also of the same section, and after the wire is passed through all the holes, and the blades have been carefully spaced, the tail of the comma between the blades is forcibly bent out by suitable pliers so as to act as spacing pieces. The blades on the two ends of the rotor are turned in opposite directions, as is clearly seen in Fig. 6, which also shows the water passages through the center of the rotor. Fig. 4 shows the inside view of the stator, the blades of which are flat.

THE ACTION

With this description of the parts, the action of the device will be easily understood. Referring to Fig. 5, the water enters the casing through the passages marked *C*. The passages marked *D* are for the escape of steam generated by the transformation of the power into heat. As said, it is not practical to carry off all the heat by evaporation alone. If this were done, it would mean the generation of 15,000 pounds of steam per hour, giving rise to the difficulties already mentioned. Water entering through the passages *C* flows into the center of the rotor, is thrown out through the central ports, which in Fig. 7 are marked *A*, and from there to each side, being caught by the rapidly revolving turbine blades of the rotor. These blades are shown in solid black in Fig. 7, while the cross-hatched areas represent the flat blades of the stator.

After being violently whirled through the first set of movable blades, the water passes through a set of stationary blades, from there to the next set of movable blades and so on, being alternately whirled around and retarded by the moving and stationary blades, until what has not already been evaporated, is discharged at each side into the semi-circular passages, from which it again flows into the center of the rotor. Here it mixes with the fresh water coming in through the inlet passages and again goes through the central ports of the rotor, out into the blades and around and around as the action continues. The surplus water flows out through the bottom, as shown by the passages *E E* in Fig. 5, and by the time it comes out of these passages it is practically at boiling temperature. The semi-circular passages are also cast with baffles, as shown in Fig. 4, to assist in retarding the rotation of the water.

In the calculation of the horsepower developed, it is, of course, first necessary to determine the zero reading, or the dead weight of the overhanging lever arm. Moreover, as the journal friction affects this very materially when standing still, a screw jack was interposed between the lever arm and the scale, and weight read-

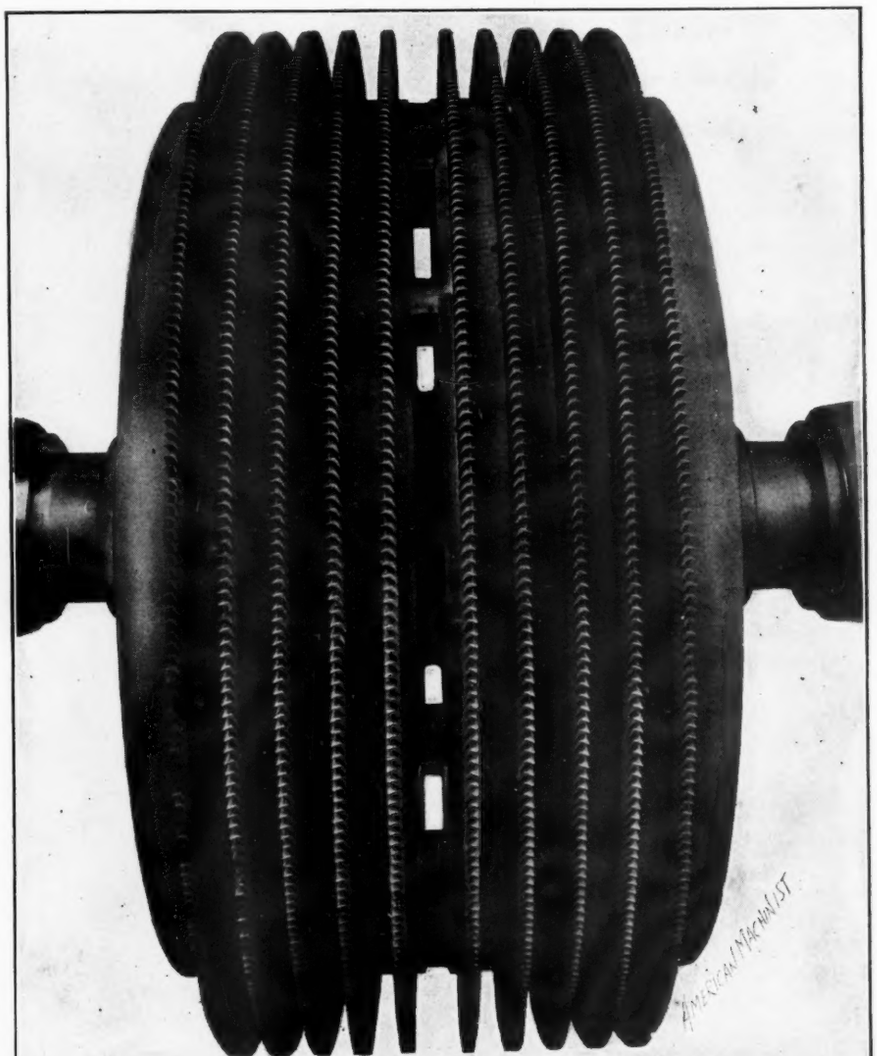


FIG. 6. VIEW OF ROTOR, SHOWING TURBINE BLADES AND CENTER WATER PASSAGES

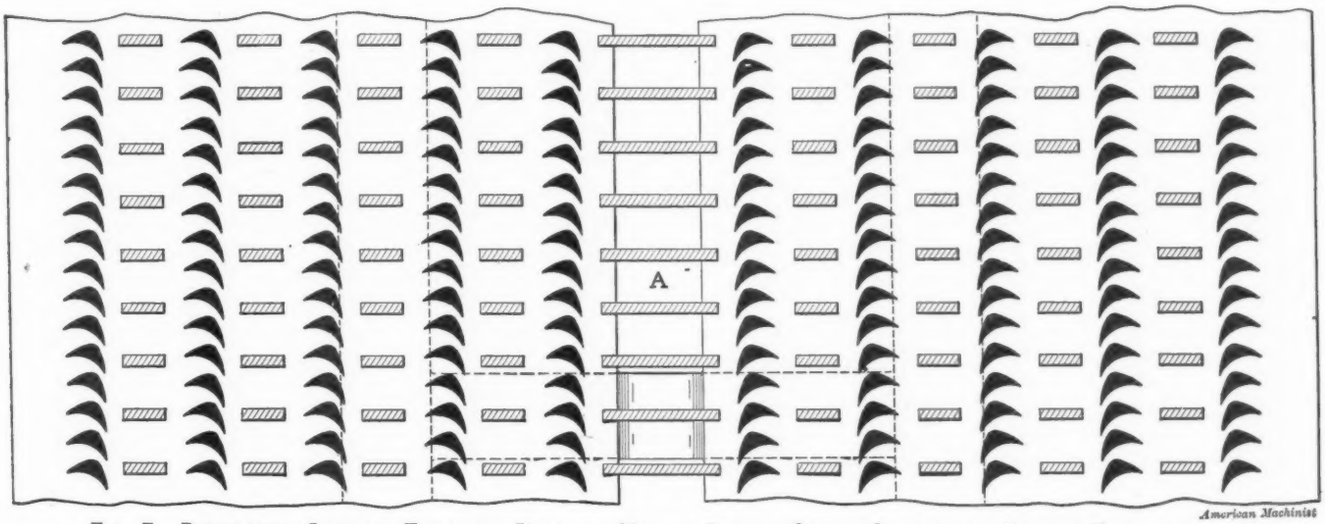


FIG. 7. DEVELOPED SECTION THROUGH BLADES. MOVING BLADES SOLID; STATIONARY BLADES CROSSHATCHED

ings were taken when screwing the jack both up and down. The mean between these two was taken as the correct zero reading.

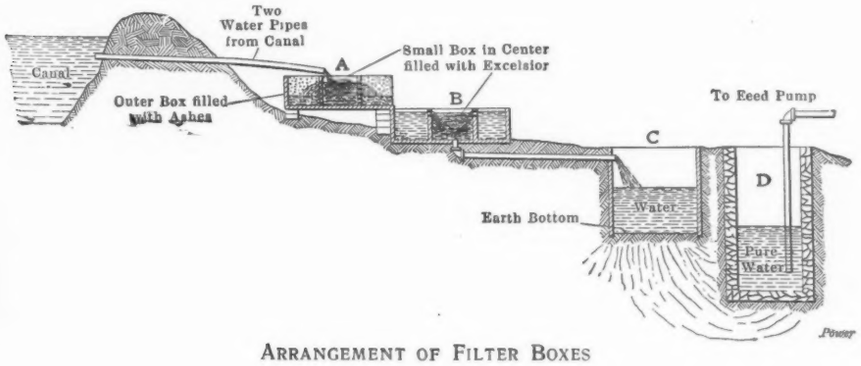
THE INPUT

So far we have considered only the measurement of the output of power. It has already been said that the object of the test was to determine the efficiency of the gearing. In order to do this, it is just as essential to know the input as to know the output. In order to determine the input for this particular set of experiments, a steam turbine, of suitable size and speed, was carried through a long series of brake tests from which it was found that for each speed and load, with a constant vacuum, and with steam of uniform quality, there is a corresponding inlet steam pressure. By varying the conditions and measuring the output of the turbine, the turbine itself was practically calibrated so that for any given set of conditions, the actual output could be accurately predicted. In this way, then, knowing the output of the turbine for a given set of conditions, and measuring the power absorbed by the dynamometer, the efficiency of the gearing was easily determined. It is said that this dynamometer has been an entire success in spite of the fact that it was radically new.

Filtering Water for Boiler Use

There are cases, especially in districts where woolen and cotton mills abound, where the dyes are emptied into the river or canals, which makes the water practically useless for any purposes except that of power. In many places the water problem is one of the chief concerns of the engineer and plant manager, and any means that will permit using water that can be obtained free of cost is of importance.

In the illustration is shown a system by which the water of a canal into which the dyes and waste matters from several manufacturing establishments were dumped, was made fit to use for boiler purposes, besides that of general use throughout the factory. The canal water, being at a higher level than the yard of the boiler and engine room, water was delivered to a box, 15 feet square, by gravity. In the center of this box was a smaller box *A* in which excelsior was placed, the small box resting upon ordinary boiler ashes, which completely filled the larger box.



ARRANGEMENT OF FILTER BOXES

The water, filtering through the excelsior and ashes, removed the bulk of the impurities of the water, and filtered into the second filtering box. In the center of this box was a smaller one *B*, in which excelsior was packed and from which a drain pipe extended to the reservoir *C*, which was made with a natural-earth bottom, but boarded up on the ends and sides. From the reservoir *C* the water seeps through the ground to the well *D*, from which it is pumped throughout the factory and also used for boiler feed.

The engineer, in telling of the results obtained by this method of filtering the almost ink-black water of the canal, stated that the boilers, before the filter system was put in operation, would foam in from two to three days after being washed out, but after washing out foaming did not occur, and it is not necessary to wash a boiler oftener than once in six or seven weeks, making not only a saving in the labor for washing boilers and coal consumption for raising steam from a cold boiler, but also saving the cost of feed water for the boilers, as well as considerable water used throughout the manufacturing plant.

The mining and metallurgical industry of Illinois in 1907 represented an output valued at over $150,000,000. The general public is, however, so accustomed to consider Illinois as an agricultural State that it does not realize that she ranks third among the States of the United States in the amount of her mineral and metallurgical output, being surpassed in this respect only by Pennsylvania and Ohio.

High and Low Water Alarms for Boilers

Descriptions of Various Types of Those Now in Use, Giving Practical Suggestions for Their Location and Connection

BY W. H. WAKEMAN

As a general rule, to which there are very few exceptions, a fireman who has always used plain water columns objects to the introduction of high- and low-water alarms in his plant, and does not desire a situation in another plant where failure to maintain a safe and economical water level is exposed without delay. Of course he claims that these safeguards are not necessary, as he will keep just the right

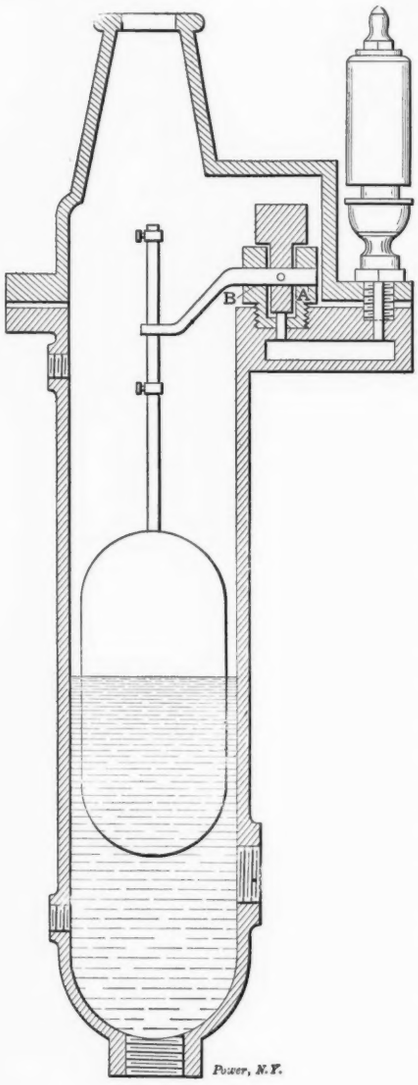


FIG. 1

quantity of water on hand at all times, but if this is done, the alarms are never brought into service, hence there can be no legitimate objection to them.

My plant is equipped with five water columns, including a high- and low-water alarm on each, and the result of my experience with new firemen is that every one of them causes these whistles to blow on account of failure to maintain a proper water level in each boiler; but as soon as they become fully convinced by experience, and not by verbal instruction, that inattention to business will surely be exposed, there is no further trouble along this line. For illustration, with five columns in use, giving ten chances constantly for whistles to blow, my present fireman does not let an alarm sound more than once a month on an average, unless there is a good reason for it. The water supply is not always fully under his control, because returns from two heating systems are fed into the boilers and when water comes from these sources faster than it can be used, it must go to the sewer direct, or be pumped into the boilers until one or more alarms are given for high water, making it necessary to blow some of it into the sewer. This seldom happens except when the plant is started in the morning.

The service in which this plant is used varies greatly, hence all of the boilers are seldom required for a whole day. When the fire under one of them is banked, it is necessary to leave the feed valve open until the furnace and combustion chamber cool more or less, as otherwise the low-water alarm would sound, on account of the continued evaporation due to this stored and liberated heat. After this has ceased, the water level slowly rises and unless the feed is shut off, the high-water alarm whistle will blow. This appears to be unnecessary and it probably is, but inasmuch as the fire is banked, less attention is given to the water level until high water causes an alarm to be given a few times and the engineer is called on to blow down the boiler; then there is no more trouble from this cause. The fireman is not allowed to open a blowoff valve. If he were he would take less pride in regulating the feed, because he could correct his mistake without reporting it to the engineer in charge.

Fig. 1 illustrates a water column fitted with an oblong float carrying a vertical rod which passes through a whistle-valve lever. This rod is fitted with two collars which can be fastened in any desired position by set screws. The illustration shows the float under normal conditions with about two gages of water in the column. The weight of the lever and valve, also the steam pressure acting on the latter, prevents the whistle from blowing. If the water level is raised, carrying the float upward until the lower collar strikes the lever and raises it, the short end rests on a fulcrum at *A* and the valve is opened, admitting steam to the

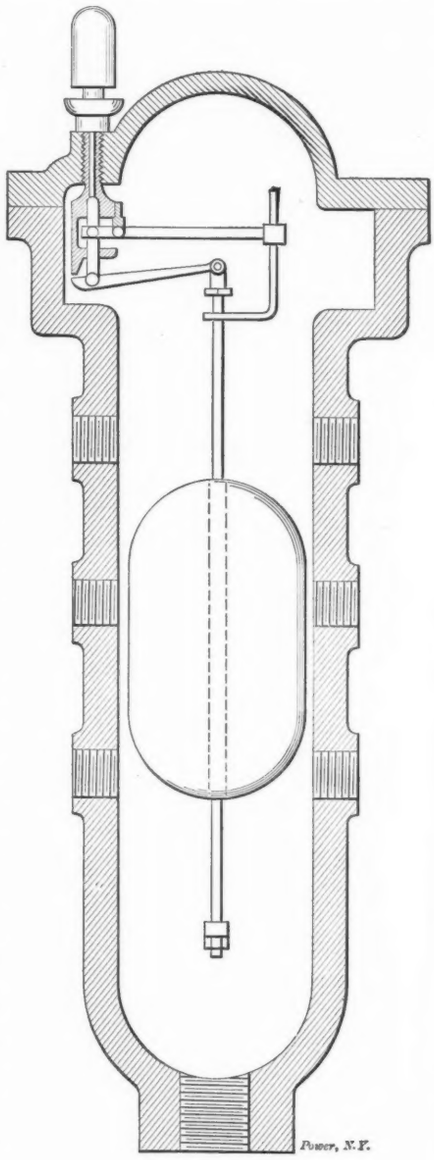


FIG. 2

whistle. If the water level falls from its present position, thus lowering the float until the upper collar strikes the lever, the fulcrum is then secured at *B* and the valve is opened as before.

Fig. 2 is fitted with one oblong float

and two levers which operate as follows: A tube passes lengthwise through the float and is neatly brazed therein at both ends. The long vertical rod is a loose fit in this tube, hence under normal conditions the float takes the position shown; but if it falls low enough to approach the danger line, the float slides down on the rod until it strikes the nuts near the end. This depresses the lower lever and blows the whistle. When the water level is raised too high, the float slides up until it strikes a curved extension of the upper lever which blows the whistle.

Observation of the operation of many plants shows that there are at least two different ways of regulating the feed wa-

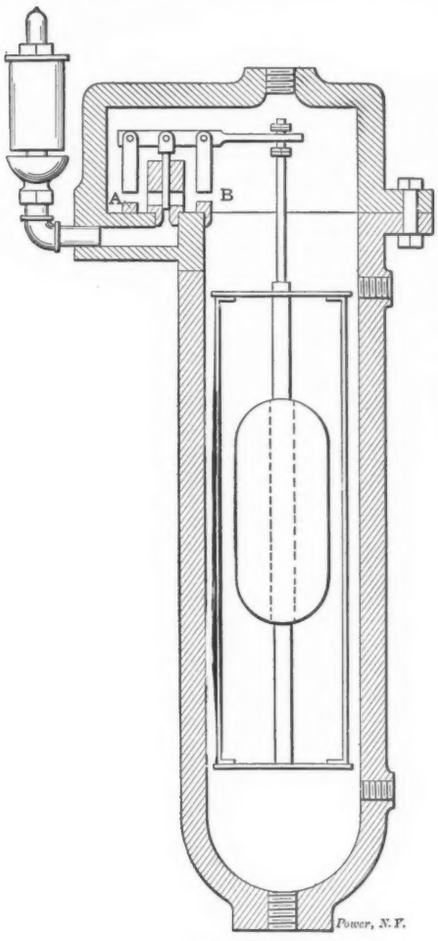


FIG. 3

ter of boilers equipped with high- and low-water alarms, as follows: One is to pump in enough to cause the high-water alarm to blow, then shut off the supply until the low-water alarm calls attention to the condition of affairs. The feed is turned on and the operation repeated. Where this plan is adopted there is danger of wrecking the engine and burning the boilers every day, for if either alarm fails to work properly when a fireman or water tender is simply waiting for it to blow without paying attention to the gage cocks or the gage glass, an accident is sure to happen in that plant sooner or later. Such men are not worried by the almost continuous blowing of one or more whistles, although every blast gives warning of danger, and their apparent indifference is evidence of carelessness or ignorance, or possibly a combination of both of these unfortunate qualifications. These remarks do not apply to those boilers which hold so little water that the water level varies rapidly, nor to those firemen and water tenders who are zealous in the discharge of their duties, but find it impossible totally to prevent the sounding of alarms. There is a vast difference between these two classes.

For illustration, take a certain case where a water-tube boiler is fitted with a gage glass about 18 inches long. If this is filled to the top and the pump stopped, it will go down to the bottom in three minutes. The boiler feed must be regulated by hand, while one man does the firing, cares for the engine, etc. In such a case it is very difficult properly to control the water level, therefore if a whistle is heard occasionally there is a good reason for it.

The other plan is to keep the water level in each boiler central between the higher and lower safe limits, or as near to this desirable point as possible. Where an alarm does sound on account of conditions that a fireman or water tender cannot control, if he has right ideas upon the subject he shows by earnest efforts to correct the evil that he is not satisfied with such results and will endeavor to secure improvements.

Members of the indifferent class never remove or muffle a whistle in order to prevent detection of their inefficient service, because it does not cause them anxiety, but when a competent and conscientious engineer in charge of one or more boilers finds that he must work under conditions which make it impracticable for him to maintain the water level in each to his satisfaction, he has been known to fix the whistles so that they will give warning in the boiler room, but will not blow loud enough to annoy the entire neighborhood, and this action is fully justified for given conditions.

Fig. 3 is fitted with a steel float coated with copper to prevent corrosion. It is protected from contact with the rough casting by a cage, only a portion of which is shown in the illustration, but the principle involved is clearly presented. When the water level is raised higher than it ought to be, the short end of the lever is supported by contact with a lug at *A*, which action opens the whistle valve. If the feed water is shut off and the water level falls below the first gage cock, the float carries the long end of this lever downward until contact is secured at *B*, thus opening the whistle valve and giving an alarm.

Fig. 4 illustrates a water column fitted with two oblong floats, each one operating a separate valve, but both of them admit steam to one whistle. While the water level is kept below the upper float it keeps the valve shut, but when the float is raised by water it blows the whistle. Lowering the water level to its proper place closes this valve and if it is further lowered until the lower float is no longer supported by it, the other valve is opened and the whistle blows again.

It is claimed by some engineers that a water column containing a high-water alarm, also one for giving warning when the water is getting low, is a dangerous

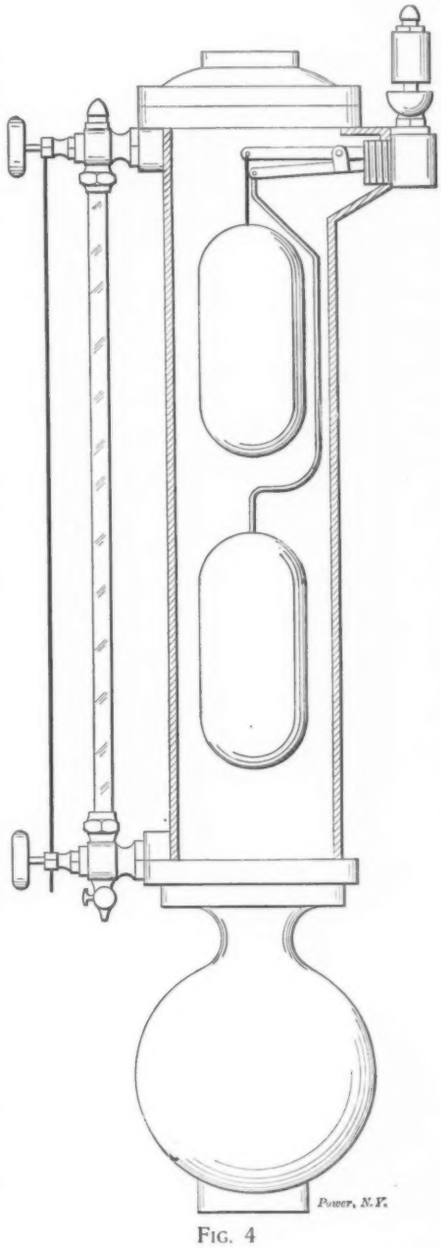


FIG. 4

appliance for the following reasons: When the fireman of a plant hears an alarm whistle blowing, he may assume that there is too much water in that boiler and proceed to open the blowoff valve, which is seldom or never located where he can do this and watch the glass water gage at the same time. After waiting a reasonable time and finding that the whistle continues to blow, he closes the blowoff valve and walks around to the front. There is no water in the glass and he does not

know how low it is, as it is below the first gage cock. He proceeds to restore a proper water level and the incident is closed, without damage to the boiler, although it might have resulted in overheated tubes or flues, if the water level had not been raised promptly.

While such a statement concerning the possible action of a fireman seems almost incredible, still it is not a fancy sketch, and as it is based on facts it demands careful consideration. However, the result is that more intelligence and commonsense should be used in operating steam boilers, rather than to reduce the number of automatic safety appliances now in use. If the fireman is not allowed to open the blowoff valves, but is obliged to report to the engineer whenever his alarm whistles blow, it compels the latter to verify the report of the former, and there is little danger of both making the same mistake at one time.

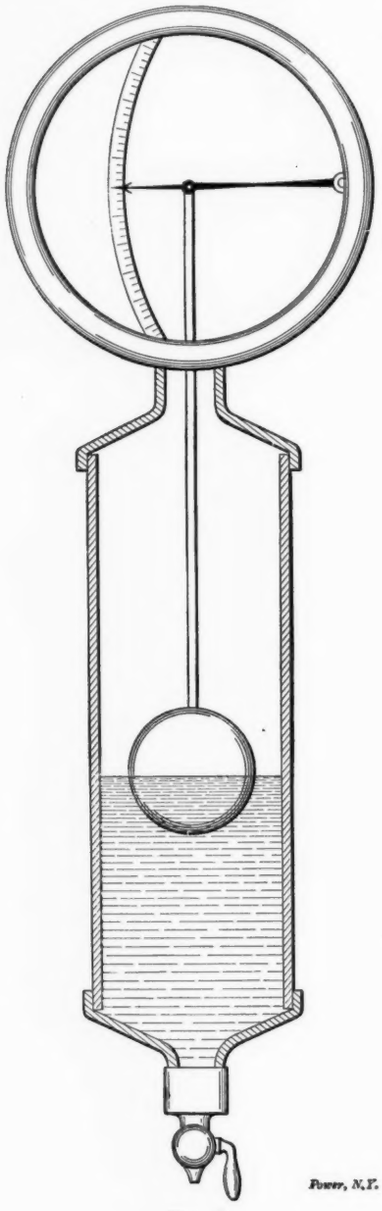


FIG. 5

In cases where only one man is employed, and he acts as both fireman and engineer, he should be competent to tell whether the water level is too high or too low, and if he is not, then it is in order to secure a more intelligent engineer without delay.

The idea of dispensing with high-water alarms is at least partly due to failure to appreciate the disadvantages of running without them. There are many boilers in use in which the engineer does not allow three gages of water while the engine is at work, because he would get too much water into the cylinder for safety. If high water in some boilers causes an excessive amount of moisture to be carried out with the steam, is it not safe to assume that even where enough does not go over to cause pounding in the cylinder, yet more is constantly thrown out than would be if less than two gages were the rule rather than the exception? Dry steam is more valuable for use in an engine than an equal weight of water and steam, hence it naturally follows that a high-water alarm will prevent waste of pure water, give more steam for a given weight of coal burned, and supply steam that is more valuable when a stated volume is considered.

If these results are secured in 99 per cent. of the boilers that are equipped with these devices, and with 1 per cent. there is danger of a stupid fireman blowing down when he ought not to, the only result of which is loss of a small amount of water and heat, is it not a businesslike proposition to risk the latter to secure the former?

The result of my experience for the past 15 years is that high- and low-water alarms need considerable attention in order to insure prompt action. Floats fill with water, small screws are lost, joints corrode and do not respond quickly, and it is not always convenient to make repairs when they are needed. Owing to existing conditions the labor cost of these repairs is paid by myself, yet when everything is considered, these appliances are worth much more than they cost, hence I continue to use them with satisfaction. When taking these columns down after shutting off steam and water by suitable valves in the connections, I sometimes wonder if those engineers who object to the use of valves here, maintain their high- and low-water alarms in good order at all times, and if so, how do they manage to lay off a boiler, whenever one of these appliances needs repairs, without disabling the plant and causing it to be shut down? If they use only plain columns without devices to give alarms, their opinions concerning the use of valves are of little value, because the requirements in such cases are less than for a well-equipped plant.

As there are several wrong ways to connect a water column with high- and low-water alarms to a boiler, the following suggestions if put into practice will enable the inexperienced engineer to avoid dangerous mistakes along this line:

Examine the boiler and compare it with the column in order to decide on the proper hight for it. There should be at least 3 inches of water over the tubes in an ordinary tubular boiler, when it first appears in the gage glass. Having adopted this as a safe rule, and assuming that the water must be still deeper when the low-water alarm ceases to blow, apply it to any other boiler that is to be used. Having decided the hight, proceed to make the top connection into the steam space.

Do not connect this into any other pipe, nor connect another pipe into it, except for a steam gage, as there is no outlet to such a connection. Arrange this pipe so that every inch of it will drain either into the column, or into the boiler. Do not have a pocket at any point, caused by either a bent pipe, a globe valve, or the use of too many fittings. Arrange the lower connection so that water from the column will drain through every inch of the pipe back into the boiler.

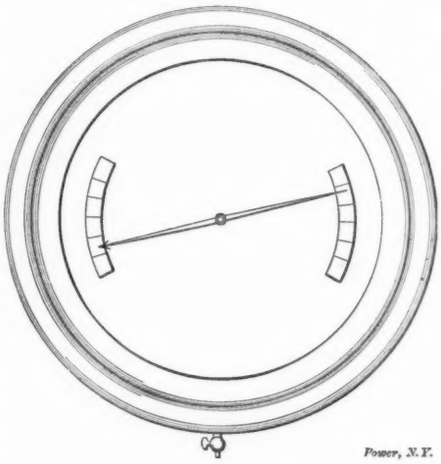


FIG. 6

Do not tap this pipe into the lower part of the shell, where scale and sediment can fall into it, but put it through the front head several inches from the shell. It is not absolutely necessary that both top and bottom connections shall be of the same size, but uniformity in this respect gives the whole job a good appearance. Use not less than 1-inch pipe for small boilers, and 1¼-inch for larger sizes. Brass pipe is suitable for this work, but if it is not available, select the extra-strong wrought-iron kind, because it is safe and durable.

For steam-heating plants where water resulting from condensation is fed back into the boilers, the lower connection may be made directly into the sediment chamber, as it makes a good-looking job, and usually saves one or more ells. If the hole is not large enough to take the pipe mentioned, it may be reamed out and properly tapped, provided there is iron

enough at this point to make it perfectly safe. Where comparatively impure water must be used, and the condensation is not returned, the lower connection should be made into the side, and a drip pipe put on the sediment chamber.

No alarm is used in connection with Fig. 5, but it indicates the hight of water at all times in the steam boiler to which it is attached. A copper float, which is filled with compressed air in order to strengthen it and prevent external pressure from forcing water into it, rests upon the surface of solid water, not being affected by foaming or priming. This float is in a detached chamber, resembling a water column, in direct communication with the boiler, thus giving the same water level in both. By means of a stem which passes up through a steam-tight frictionless joint and connects with a light lever or pointer, the hight of water is read the same as the pressure of steam on an ordinary gage.

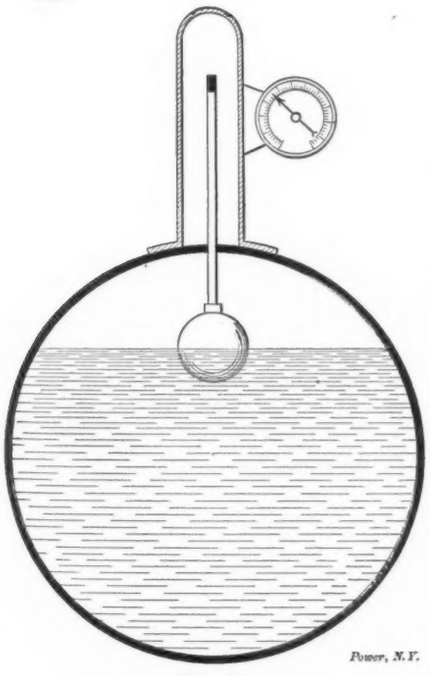


FIG. 8

Fig. 6 is another water gage which appears like a steam gage and the water level is indicated by a hand or pointer. In this case, however, there is no mechanical connection between the float on the surface of the water in the boiler and this pointer, as magnetism is the agent employed for this purpose. Fig. 7 shows the float attached to a stem bent at right angles which carries the magnetized arms at the right hand, one of which is a positive and the other a negative pole. The gage is shown in section, giving a side view of the needle or pointer on the outside. The stem, which is made of brass, moves freely, as there is no stuffing box to cause friction. There are 6 degrees indicated on the dial, Fig. 6, and as each degree represents a rise or fall of 2 inches of the float, the complete gage indicates a maximum difference in the water level of 12 inches. The theory of such a water gage is correct, and it will work perfectly as long as the float remains in perfect order, with the magnetized arms fully charged, thus duly attracting the needle.

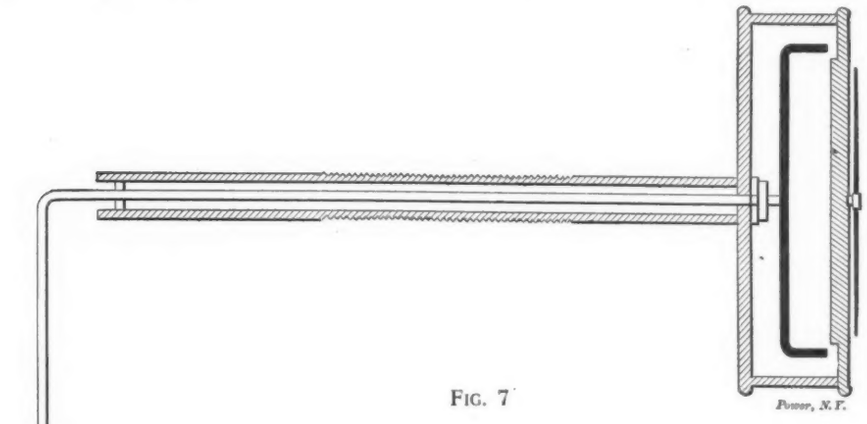


FIG. 7

Fig. 8 illustrates another magnetic water gage, but it has a float and straight vertical stem entirely within the boiler, hence there is no stuffing box or other joint to be kept steam-tight and thus cause friction that will interfere with the free operation of the device. The upper end of this stem is a magnet, hence it attracts the pointer and the true water is seen at a glance.

Fig. 9 illustrates a device for transmitting the reading of an ordinary water gage, located on a high vertical boiler down to a hight of, say, 6 feet above the boiler-room floor, thus enabling the fireman to note his water level at a glance, without depending on what he apparently sees at a distance of 20 feet more or less. When viewed at this distance the resulting angle causes the water level as indicated by the glass to appear higher than it really is; hence if the fireman fails to take this into acount he may conclude that he has 1½ gages of water when really his low-water alarm is about to blow. With a water gage at or very near the hight of the fireman, he sees the indicator in its proper relation to the boiler and regulates the feed water accordingly.

This desirable result is secured by connecting another water column to the regular combination, as shown in the illustration, by 1¼-inch nipples. This column has no water gage nor trycocks, but it contains a copper float which is partially submerged to a uniform depth at all times. A 3/4-inch pipe is attached to the bottom of the auxiliary column and extends downward, as required. Its lower end is connected into an ordinary water gage, the lower connection of which is fitted with a pet cock for blowing out sediment, as this gage and the pipe are constantly full of water. An aluminum rod is attached to the float, and the end of it is painted red that it may be easily seen. It occupies the same relative position in the auxiliary water-gage glass that the true water level maintains in the upper glass, hence is a reliable indicator of the amount of water in the boiler.

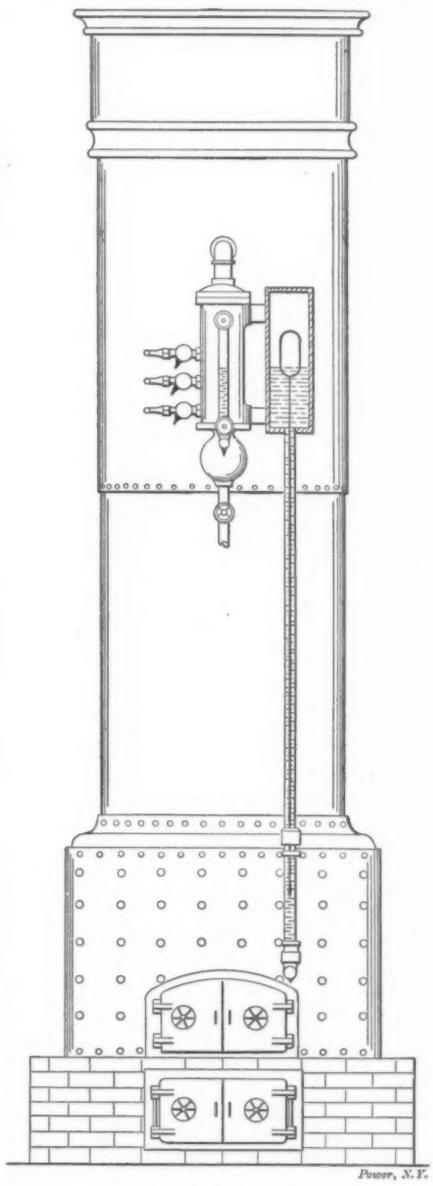


FIG. 9

ELECTRICAL DEPARTMENT

Especially conducted to be of Interest and Service to the men in charge of the Electrical Equipment

Electric Elevators

BY WILLIAM BAXTER, JR.

One of the latest types of mechanical-control electric elevator engines made by the Otis Company, and the one that is generally installed at the present time, is shown in Fig. 75. The principal difference in the machine proper between this design and the one described in the preceding article is in the brake, which is of the magnetic type. In the control apparatus there is a greater difference, as

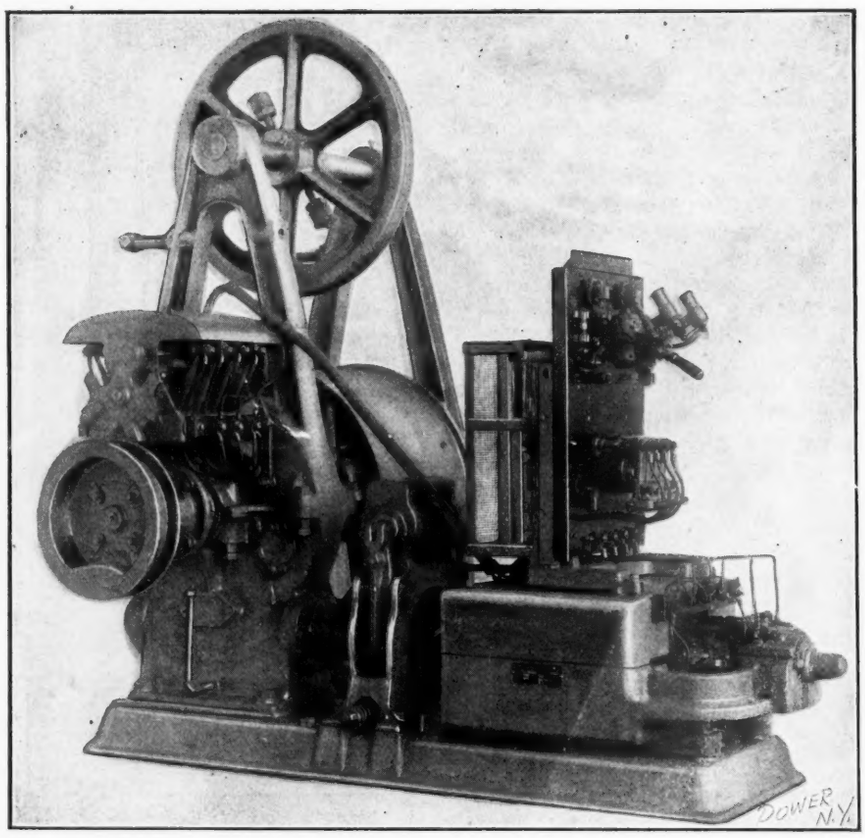


FIG. 75. ONE OF THE LATEST TYPES OF OTIS MACHINE

can be clearly seen from the illustration. The reversing switch and the other parts of the controller are not contained within a single box, but if they were the box would be of much larger size than the one used with the earlier type of machine. When electric elevators first came into use all the controlling switches were made much smaller than at the present time, but experience has shown that the wear these parts are subjected to is such that they must be made decidedly substantial to endure, and the construction must be such that every part can be easily reached whenever necessary. It is for these reasons that the control apparatus in this type is more spread out than in the earlier design. The reversing switch is located in Fig. 75 directly above the stop-motion mechanism, the object of so placing it being that it might be where it could be actuated by the stop motion to stop the car at the upper and lower limits of travel. The remainder of the controlling apparatus could be placed in any convenient location, but for the sake of compactness it is mounted on top of the motor.

The slack cable stop used with this machine is of the electric type and is placed under the drum, so as to be actuated whenever the ropes sag through becoming slack. The position of the slack cable bar is seen in the line drawing, Fig. 76, which is an elevation of the machine taken from the motor end. The slack-cable switch is shown in the closed position at *E*, and is of the same design as the one that was shown in Fig. 64*. The stop-motion mechanism is mounted upon the end of the drum shaft *S*, and the operating sheave is secured to its outer end; neither part is rigidly secured to the shaft, but is carried around when the traveling nut reaches either one of the stationary nuts, in the manner already fully explained in connection with the early type of elevator machine. In Fig. 76 the stop motion is not shown but its construction can be fully understood from Fig. 77. In this drawing it will be seen that the operating sheave *A* is bolted to the end of the stop motion frame *B* and that the latter is held from running off the shaft *S* by the stop nut *N'*. The end of the shaft *S* projects through the outer end of the frame *B*, so as to give the latter support and keep it in the central position, but the frame can revolve freely on the shaft *S*. As is shown in Fig. 75, the operating sheave is cast with an enlargement of the rim at the point that rests at the bottom when the stop motion is in the running position, this construction being used so that the weight in addition to the friction of the apparatus may act to hold the frame *B* from being turned around by the friction between it and the shaft. When the traveling nut *N* strikes the nut *N'* at the inner end, or the nut *N''* at the outer end, of the frame *B*, the stop motion will be rotated, because the latter nuts are both rigidly secured to the shaft *S*. As soon as the frame *B* rotates, it will turn the shaft *B''* since the gear *B'* is in mesh with the gear on the inner end of the frame *B*. The shaft *B''* carries on its outer end a disk *C* that opens the reversing switch when moved by the stop motion in the manner just explained, and stops the elevator.

The construction of the reversing switch, and the way in which it is operated by the rotation of the stop motion frame *B* can be made clear by reference to Figs. 77, 78 and 79. Fig. 79 is a

*Page 998; issue of December 14.

photographic view of the complete switch, and on it the various parts are marked the same as in the other illustrations for convenience in cross reference. The operator in the car controls the movement of the switch by rotating the operating sheave *A* either by the direct pull of a hand rope or through a "running rope" rig which will be explained later. To start the car upward the switch sheave is rotated in one direction, and to start the car downward it is rotated in the opposite direction; to stop, the sheave is rotated back to the central position. Whenever the stop motion comes into action at either end of the car travel it rotates the sheave in the direction that it would be rotated by the operator, if he desired to stop the car; in both cases the movement is effected through the rotation of the stop-motion frame *B*. When this frame is rotated, the shaft *B″* is rotated, and the stud *C″*, carried by the disk *C*, runs a groove in the disk *D* on the shaft *D′* and thereby this shaft is turned. Upon this shaft there are four cams marked *I*, *II*, *III*, *IV*, and these act to close switch levers that connect the motor circuits and the main line in the proper order to give the direction of car travel corresponding with the direction in which the sheave *A* may have been turned. The two cams *III* and *IV* operate contact levers that constitute the reversing switch proper, the other two cams move switches that close the main circuit, and also make the proper connections with the motor, the brake and the controller-switch magnets. These several connections will be fully explained in describing the wiring diagram.

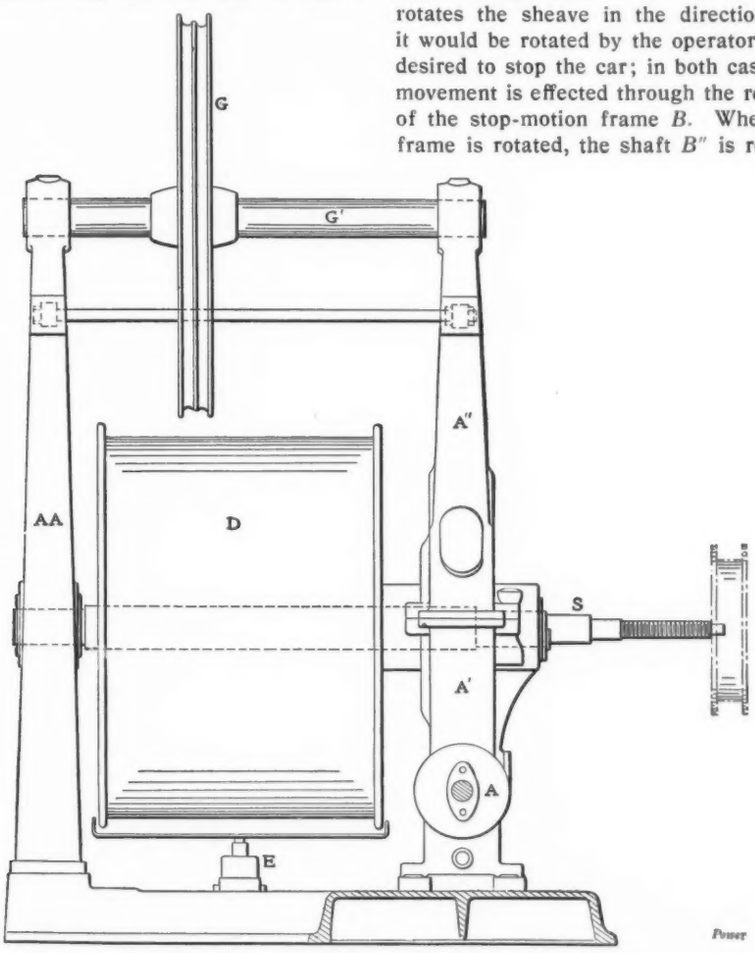


FIG. 76. HOISTING DRUM AND SLACK-CABLE STOP

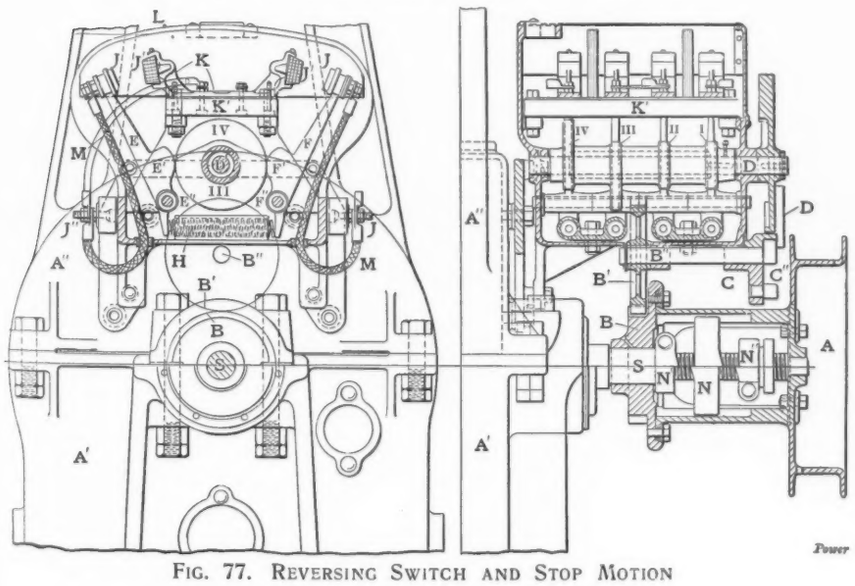


FIG. 77. REVERSING SWITCH AND STOP MOTION

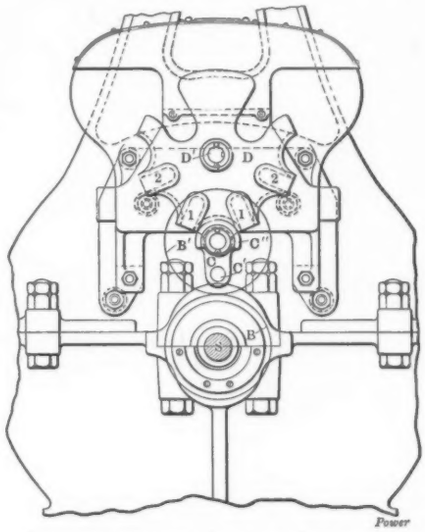


FIG. 78. OTHER END OF REVERSING SWITCH

The way in which the cams on the shaft *D′* move the switch levers can be understood from the end view in Fig. 77, where the cam *III* is shown in full outline and the cam *IV* occupies the upper half of the circle. Both these cams are of the same shape but are set on the shaft diametrically opposite; therefore, if the shaft is rotated clockwise, the cam *III* will permit the lever *F* to be drawn to the left by the spring *H* so as to close the contacts *J* and *J′*; at the same time the cam *IV*, which is just back of the cam *III*, will permit another spring *H* to

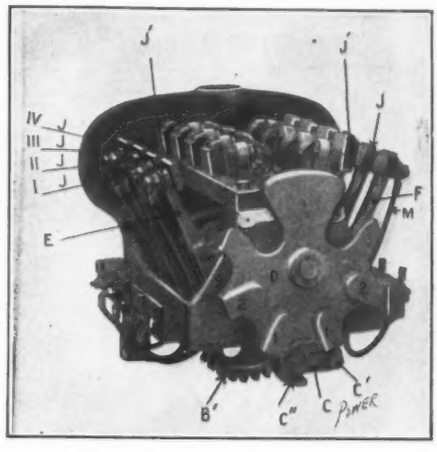


FIG. 79. REVERSING SWITCH

draw its lever *E* to the right so as to close its contacts *J* and *J'*. If the shaft *D'* were turned in the opposite direction the switches would be closed in the opposite order; that is, the switch *E* opposite the cam *III* and the switch *F* opposite the cam *IV* would be closed. The cams *I* and *II* are made so that when rotated in either direction they permit both the switch levers opposite them to be closed, but one cam is made so that the switches it controls are closed sooner than the other two.

The way in which the rotation of the shaft *B''* acts through the disk *C* and the stud *C'* to rotate the shaft *D'* is shown in Fig. 78. When the disk *C* is in the position in which it is shown the rim *C''* fits in the curved surface of the disk *D* and prevents the latter from moving. When the disk *C* is rotated far enough, the rim *C''* passes out of engagement with the disk *D* and at the same instant the stud *C'* enters one of the grooves *l* and as *C* rotates, *D* is carried around. For each revolution of the disk *C* the disk *D* is rotated one step. As the distance in which an elevator can be stopped increases with the car speed, it is necessary to arrange the stop motion so that the reversing switches may be operated while the shaft *S* is rotating through the proper angular distance, and this is accomplished by varying the relative sizes of the gears *B* and *B'*. For a low speed the diameter of *B'* is reduced and for high speed it is increased; in the first case a smaller rotation of the shaft *S* will operate the switches than will be required in the second case. The current is carried to the switch-lever contacts by the cables *M*. The contacts on the ends of the levers are insulated and are made of brass. The contacts *J'* are made of carbon and are held on an insulating base *K'*.

Relays in Distribution Systems

BY J. W. HIMMELSBACH

In these days a good many power-plant operations are performed by automatic appliances which were formerly taken care of by the men operating the system. Now protective apparatus and automatic appliances have been brought to a point where a switchboard is almost entirely automatic. Among the most important automatic devices are those for relieving a system of heavy short-circuits or dangerous overloads. On low-potential systems of moderate maximum load, the familiar automatic circuit-breaker, operated by a magnet, is used. In larger plants, however, generating alternating current, the safety circuit-breaker is operated by a separate primary device known as a relay, the several types of which are explained in this article.

There are two principal types of relay, the overload relay and the reverse-current relay. Each of these two may be divided into three classes, namely: Instantaneous, time-element and inverse-time-element.

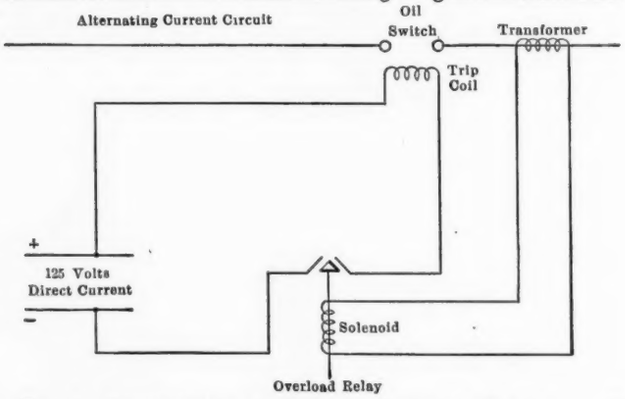


FIG. 1. CONNECTIONS OF SIMPLE OVERLOAD RELAY

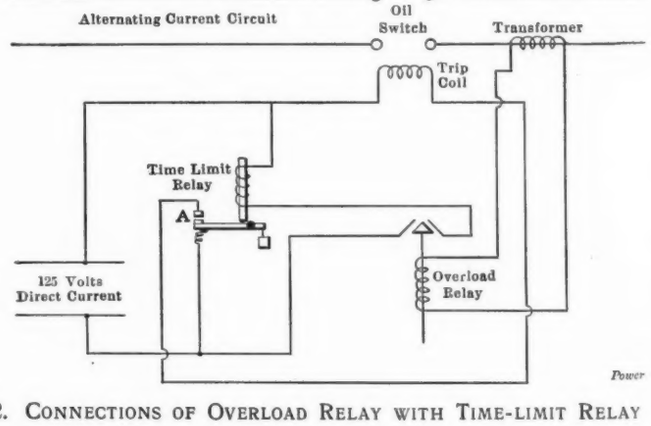


FIG. 2. CONNECTIONS OF OVERLOAD RELAY WITH TIME-LIMIT RELAY

INSTANTANEOUS OVERLOAD RELAY

When it is desired to provide for the immediate and instantaneous opening of a circuit when the load goes beyond a certain predetermined point, this type of relay is used. It must operate surely and positively, without any hesitancy, and the time taken for operation must be just as short as possible. The means of accomplishing this usually consists of a solenoid arranged to close a pair of contacts which complete the magnetizing circuit of the actual circuit-breaker, as illustrated diagrammatically in Fig. 1. In alternating-current plants the relays are supplied with current through small transformers, which permits the relay to be made small and compact, and the use of the same winding for all capacities. The secondary windings of these transformers are usually designed for a full-load current of five amperes, except in special cases, and the relay solenoid is designed to operate at that current, or whatever the full-load current of the transformer is. The relay is constructed so that the position of the plunger can be altered, and thereby made to operate at different strengths of current in the solenoid. A graduated scale is provided for setting the plunger to move at the desired current flow.

Referring to the diagram of connections in Fig. 1, the operation of the relay is as follows: Supposing, for example, that the greatest allowable current in the alternating-current circuit is 250 amperes, the ratio of the relay transformer will be 250 to 5; therefore, with 250 amperes flowing in the main circuit, the current in the secondary circuit of the transformer and the relay coil will be 5 amperes. The plunger must then be set so it will pull up and close the contacts when the current through the solenoid becomes a fraction greater than 5 amperes—say 5.1 amperes, for example. Under these conditions, if the current in the main circuit should increase to 255 amperes, the relay current will be 5.1 amperes, and the plunger will be lifted, closing the direct-current circuit through the circuit-breaker magnet coil, and causing it to open the oil switch in the main circuit. This stops the current in the small transformer, and the relay plunger falls back into the "open" position, ready for the next time it may be called upon. The oil switch

remains open until closed again by the switchboard man.

This is the simplest form of overload relay. It will be evident that even when the load on the main circuit is only 50 amperes, a short-circuit lasting for only a few seconds will operate the relay and open the circuit. In order to get rid of this disadvantage the overload relay with an adjustable time element has been devised.

Time-element Overload Relay

This instrument is practically the overload relay fitted with a contrivance which will delay its operation. The time-element attachment is such that the relay will not be operated by momentary excessive overloads or short-circuits, but will, nevertheless, be ready to operate if the overload lasts longer than a certain set time. This can be accomplished in two ways, either by making the overload relay with an attachment for delaying the action of the solenoid, or by adding a second relay to give the time-element feature. The relay manufactured by the General Electric Company will serve as an example of the first type. In this relay the plunger works against an air bellows which has a vent for the escape of the air under pressure. The size of the opening is adjustable by means of a set screw, and in that way allows the retarding effect of the bellows to be increased or decreased at will. This device gives very excellent results, and by entirely removing the set screw, which gives the maximum size to the opening, the action of the relay can be made almost instantaneous. The maximum delay obtainable is about 15 seconds. The operation and connections for this type of relay are similar to those for the instantaneous relay, shown in Fig. 1, the retarded motion of the plunger being the only point of difference.

The arrangement with a second or auxiliary relay is well illustrated by the apparatus made by the Westinghouse Electric and Manufacturing Company, in which an instantaneous overload relay is used in connection with a separate retarding relay. The overload relay is arranged to close the operating circuit of the time-element relay, which in turn closes the tripping circuit of the oil switch. The overload relay is built on the principle already described, except that the contacts are much lighter in construction, because only the operating current for the time-element relay passes through them. In the Westinghouse overload relay the plunger, instead of being pulled up, as represented in the diagram, Fig. 2, is pulled down against the action of counterbalancing weights, which can be changed to adjust the closing point. The time relay consists of a pivoted arm, weighted at one end, and carrying a contact *A* on the other end, a solenoid, and a plunger so arranged that when the solenoid is energized it lifts the plunger and allows the contact end of the arm to be pulled up by the weight on the other end of the arm. The movement of the arm is controlled by an air dashpot equipped with a graphite plunger. The time element is varied by varying the distance through which the arm must travel before making the contact which closes the tripping circuit of the oil switch.

The operation of the apparatus is similar to that of the simple overload relay. When the current in the solenoid of the overload relay exceeds that for which it is set, the plunger is drawn up and the circuit is closed through the solenoid of the time-limit relay, which fits its plunger immediately and allows the arm to rise slowly until the contacts *A* close; this closes the circuit of the oil-switch tripping magnet. It is to be noticed that in this form of relay the time element is absolutely independent of the amount or character of the overload; if the time limit is set for three seconds the relay will operate in three seconds, whether a heavy short-circuit or a slight overload is the cause of its operation. The disadvantages of this type of time-element relay are the cost, on account of there being two relays, and the requirement of more space for mounting than the bellows type, because it is not self-contained. It has the great advantage, however, of operating at the time interval for which it is set, regardless of the degree of overload, as just explained.

As mentioned in the beginning, some time-element relays have a time limit that is longer for slight overloads than for heavy overloads. That type is known as the

Inverse Time-element Overload Relay

As the name implies, the time element in this relay varies inversely as the degree of overload which causes it to operate. For example, if the inverse time-element relay were set to operate in three seconds at a load of 250 amperes, the relay, with precisely the same setting, would operate in, say, two seconds at a load of 500 amperes.

The time-element relay retarded by a bellows has this feature incorporated in its design, but the results are only partially successful. I have found from experience that when this relay is subjected to a very severe short-circuit, instead of operating almost instantaneously it has a tendency to hang up and not operate as quickly as it would under a less severe overload. This is probably due to the excessive vibration caused by the heavy load and the resulting friction of the plunger against the solenoid. Moreover, the vent designed to relieve the air pressure in the bellows is so small that the sudden pressure in the bellows cannot be relieved quickly enough to allow the relay to perform properly. However, the faultiness of its operation is not very difficult to overcome.

So far only single-pole relays have been discussed, and it is hardly necessary to go into details about relays used for the protection of polyphase circuits, as the design and operation of a polyphase relay is precisely the same as that for a single-phase relay, except for the fact that the polyphase relay is a two-pole apparatus consisting of two single-pole relays mounted on the same base with contacts so connected that the closing of

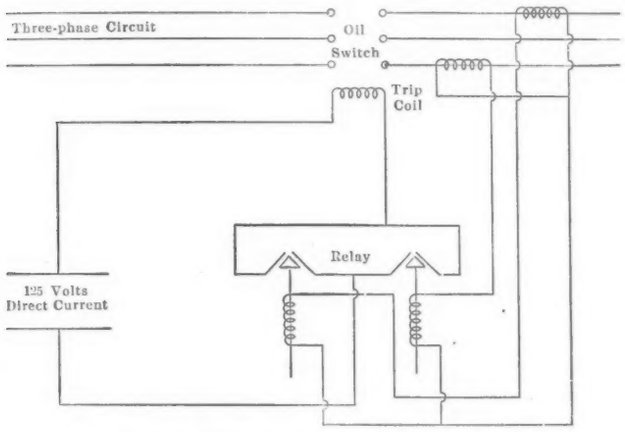


Fig. 3. Three-phase Overload Relay with Two Transformers

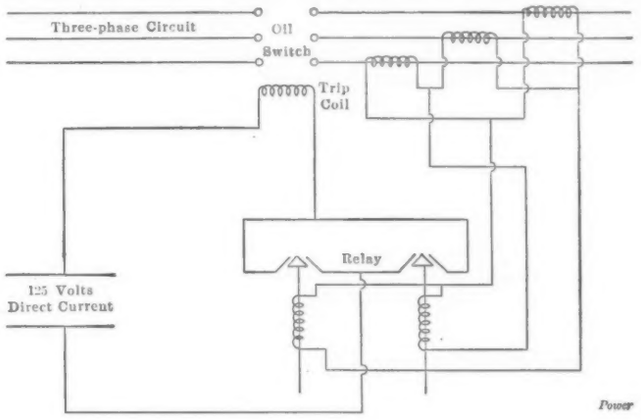


Fig. 4. Three-phase Overload Relay with Three Transformers

either one will close the tripping circuit of the oil switch. However, it might be well to add a word in regard to the current-transformer connections when using a polyphase relay. In a two-phase system, two transformers are used, one in each division of the circuit, and the connections are precisely the same as they would be for two single-phase circuits. For a three-phase system, two transformers connected V-fashion may be used; but the use of three transformers connected in delta with one phase reversed is preferable, for the reason that an overload on any of the three phases will operate the relay. Fig. 3 shows the connections for two current transformers and Fig. 4 shows the connections when three transformers are used.

General Application

Overload relays are used on all lines where it is desirable that they should be protected from severe overloads or short-circuits. This applies to outgoing feeders from a central generating station, incoming feeders of a substation and distributing feeders from a substation. A plan which affords perhaps the best protection to a system is to protect the generating-station end of a feeder with a time-limit relay set for four seconds, the incoming substation feeders with an inverse time-element relay set for two seconds and the distributing feeders leading out from the substation with an instantaneous overload relay. The advantage of this plan lies in the fact that, if any disturbance occurs on a distributing feeder, the relay for that feeder alone will operate first, as it acts immediately, thereby keeping the service between the generating station and substation intact; but if trouble should appear which affects the whole substation and which is not relieved by the operation of the distributing-feeder relays, the relays at the substation end of the main feeders, being set to act in a shorter time than those at the generating end, will operate, cutting out that substation only, and leaving undisturbed any other substation that might be connected to that feeder. Any serious trouble between generating and substations will, of course, be taken care of by the relays at the generating station.

Reverse-current Relays

Overload relays should never be installed on generator circuits, as the feeder relays are well capable of relieving the generator from any strain. However, it is common practice to provide protection to the generators and substation feeders from "reverse currents" when rotary converters are operated. For this purpose "reverse-current" relays are used. These are relays designed to open oil switches when the transfer of power is reversed; this reversal of power transference is erroneously referred to as "reverse current." When it occurs it usually comes with a rush, as it is commonly caused by the failure of some piece of apparatus, such as the stoppage of the steam supply at the generating station, or the breaking down of a prime-mover driving a generator operating in parallel with others, or the running away of a rotary converter. Under these circumstances the relays are generally manufactured with little or no time element for the reason that in almost every case quick action is desired.

Reverse-current relays are divided into two types: the differential solenoid type and the wattmeter type. For the sake of simplicity the single-pole differential relay will be described. The chief point of difference between a differential reverse-current relay and an overload relay lies in the winding of the solenoid. The solenoid for the reverse-current type is provided with two windings, a series or current winding and a shunt or potential winding. The coils are so connected that the magnetizing effect of one coil is opposed to that of the other, and they are proportioned so that under normal conditions they neutralize each other to such a degree that there is not sufficient magnetism to start the plunger. Both current and potential transformers are necessary for the operation of the relay. The conections for it are shown in Fig. 5; here the two windings on the relay are shown separated, but in the actual solenoid one is wound over the other to concentrate their magnetic effects. Under normal conditions neither the series nor shunt winding of the solenoid alone is powerful enough to operate the relay, and, of course, it is impossible for the two to operate it when both windings are excited normally because the two are connected in opposition, as described. Now, if there is a reversal of power in the line, the effect in the series coil will be exactly the same as though direct current were flowing through it and the current had been reversed. The magnetizing effects of the two windings are now alike instead of opposed, and the combined pull of the two will lift the plunger and operate the oil switch.

This type of relay will also operate in case of a very great overload or a short-circuit, because under such condition the magnetizing effect of the series winding so far exceeds that of the potential coil as to lift the plunger in spite of the opposition of the potential coil. From this it will be evident that the relay will be operated by either the potential or the series coil alone, provided that either the voltage or the current exceeds a certain value.

When a relay of this type is installed great care must be taken to insure the correct relative connections of the current and potential coils, as it is a very simple matter to connect the two coils of the relay so that the effects are alike instead of opposed when the current is in the normal direction. This relay is also made in a two-pole type, for the protection of polyphase circuits, comprising two single-pole relays mounted on one base, with the contacts so connected that the closing of either one will trip the oil switch. In connecting a two-pole relay great care must be taken that the current and potential for one solenoid are taken from the same phase, otherwise serious complications may arise. Care must also be taken, as with the single-pole type, regarding the relative connections of the potential and current windings. Fig. 6 shows the connections for a two-pole relay on a three-phase line. This diagram shows two potential transformers and three current transformers, with the current transformers connected in delta with one phase reversed. When it is realized that there

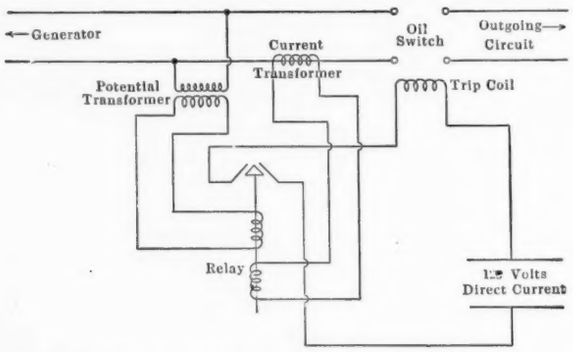


Fig. 5. Connections of Single-phase Differential Reverse-current Relay

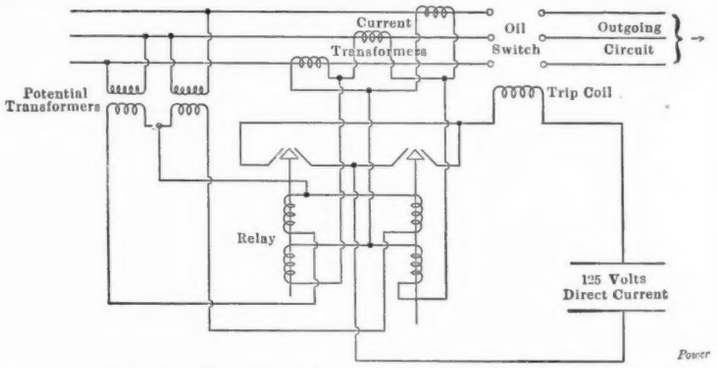


Fig. 6. Connections of Three-phase Differential Reverse-current Relay

are 36 possible methods of connecting a polyphase reverse-current relay, and that only one of these 36 combinations is correct, it is obvious that great care must be taken in installing such a relay. It may be well to mention the fact that when this relay is operated by excessive current with very low potential the operation

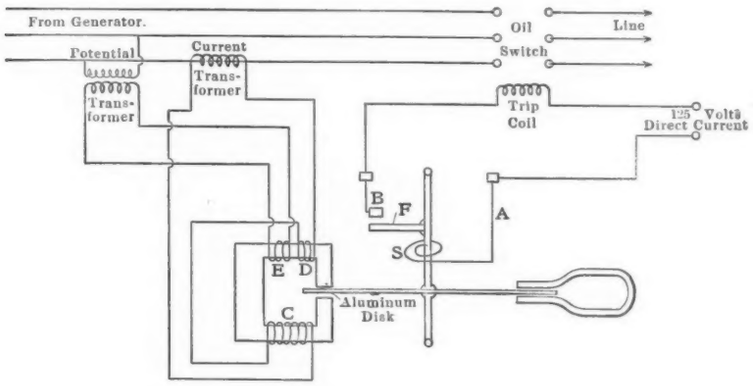


FIG. 7. DIAGRAM OF NONREVOLVING WATTMETER-TYPE REVERSE-CURRENT RELAY

can be caused by either overload or reverse current, as the magnetizing force of the potential coil is so small as to be negligible and the plunger is lifted by the series coil alone, regardless of the current direction.

WATTMETER TYPE OF REVERSE-CURRENT RELAY

If the connections of the series coil of an ordinary single-phase recording watthour meter be reversed, the direction of rotation will also be reversed. It is this characteristic of the instrument which underlies the design of the wattmeter type of reverse-current relay, and such an instrument fitted up with a contact-closing device which would be operated by the reversal of rotation would ordinarily be ready for use. But when there is no potential on the shunt winding of a wattmeter, rotation will cease, and as it is desirable to have the relay operate on current alone, some changes must be made which will cause heavy overloads or short-circuits to reverse the direction of rotation, even if the potential winding should receive no current. This is accomplished by adding a series winding to the potential coil. Even with no potential, a torque sufficient to reverse the rotation of the relay will be produced by the combined magnetic effect of the two current windings. By properly proportioning the number of turns on the auxiliary current winding, the relative effects of normal and reverse currents can be made such that the latter will operate the relay and the former will not.

There are two types of wattmeter relay, one in which the movement of the revolving element is limited to less than a single revolution, and one in which the moving element is free, under normal conditions, to revolve continuously in one direction, and while so revolving cannot operate the main switch. Fig. 7 illustrates the first type diagrammatically. An aluminum disk, mounted on a vertical spindle, is located with one edge between the poles of an electromagnet and the opposite edge between the poles of a pair of permanent magnets (only one is shown in the diagram). The electromagnet is provided with a main series or "current" winding *C*, a potential winding *E*, and an auxiliary current winding *D* to serve instead of the potential winding when there is too little potential to excite that winding properly. The spindle carries a contact finger *F*, which is held away from the stationary contact *B*, and against a stop, not shown, by a spiral spring *S*. As long as circuit conditions are normal, the magnetic fields produced by the windings *C*, *E* and *D* tend to rotate the disk backward—that is, in the same direction that the spring *S* pulls it—but the stop prevents motion in that direction. Should the direction of energy transmission in the main line be reversed, the reaction between the windings *C* and *E* will be reversed and the disk will be rotated against the pull of the spring *S*, bringing the finger *F* against the contact *B* and thereby closing the tripping circuit of the oil switch. The tension of the spring *S* is adjustable, and by this means the relay can be made to operate under any particular load condition, within certain limits. Its time element may be also varied, within a restricted range, by changing the position of the contact *B* so as to reduce or increase the distance through which finger *F* must travel in order to close the tripping circuit.

The revolving type of wattmeter relay is provided with magnets and a disk armature like those just described, but the contact-making mechanism is vitally different. The spindle carries a ratchet wheel and revolves continuously in one direction as long as the circuit conditions are normal. When it is revolving thus, a pawl mounted on a pivoted contact arm, as represented in Fig. 8, trails idly over the ratchet teeth. Should the disk reverse its direction of rotation, the pawl would catch the nearest ratchet tooth and the motion of the ratchet would force the pawl downward, and with it the pivoted arm *A*, bringing the arm against the contact pin *G* and closing the circuit of the oil-switch tripping magnet. The circles *J* and *K* represent the terminals

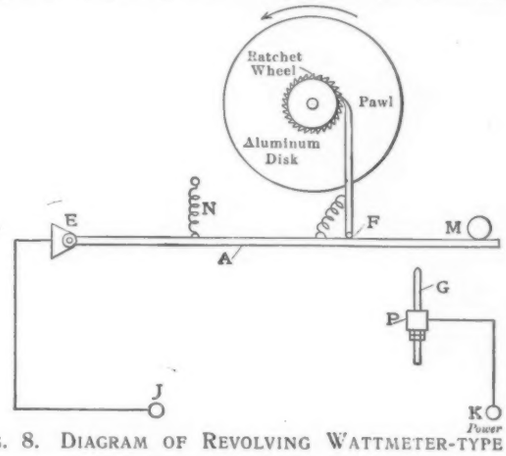


FIG. 8. DIAGRAM OF REVOLVING WATTMETER-TYPE REVERSE-CURRENT RELAY

of this magnet. The time element of the relay is adjusted by varying the distance between the arm *A* and the contact pin *G*, the latter being adjustable in the block *P*. The load setting is adjusted by varying the tension on the spring *N*.

This type of relay is very successful and is capable of very fine adjustments, as the time and load settings are absolutely independent of each other. It also has the advantage of giving visible evidence of being in operative condition, for as long as there is any current flowing at all the relay disk will rotate.

In both types of wattmeter relay it is advisable to use an auxiliary relay in the tripping circuit, as the contact mechanism is necessarily rather delicate, and on account of that fact it is not practicable to have it carry the full tripping current.

The reverse-current relay is essential on alternating-current systems to safeguard against trouble arising under the following conditions:

1. Improper field adjustment or demagnetization of the field on one of several generators operating in parallel, in which case the machine would produce a partial short-circuit on the busbars.

2. Interruption of power on transmission lines supplying rotary converters or synchronous motor-generator sets operating in parallel with storage batteries or other generators, in which event the machines in question would tend to run inverted and supply power to the transmission line.

3. Short-circuit or ground on one of several lines operating in parallel to supply power from a generating to a substation. With such a state of affairs existing, the flow of current would be from the busbars instead of to them, without the intervention of the relay.

Primer of Electricity

The Electromagnet

In the last primer lesson the relation between ampere-turns in the exciting coil of a magnet and magnetic-flux density in the magnetic circuit was explained, but no information was given as to how anyone could find out just how many ampere-turns would be needed to produce a given density. This varies so much with different grades of iron and steel that no general rule or formula will fit the case. With one kind of iron 100 ampere-turns per inch length of magnetic circuit will produce 100,000 lines per square inch area of core cross-section, while with another grade exactly the same exciting force will produce a density of only 50,000 lines per square inch. Engineers who have to do with the design and manufacture of electromagnets make careful tests of the iron and steel that they use and plot the results of these tests in the form of a chart for convenience in designing. This is not a part of an operating engineer's work under any conditions, and it is not necessary to go into the process of testing iron and steel. For practice in applying the principles of the electromagnet, however, a chart of the kind referred to is presented herewith.

Excitation Required by Iron and Steel

Fig. 20 is a chart showing the number of ampere-turns per inch of path length that must be provided to obtain a given density in average grades of annealed sheet steel, soft wrought iron and cast steel, and a good quality of close-grained cast iron. As densities are nearly always up in the thousands, it is usually convenient to express them in kilolines per square inch; kilolines are simply lines of force ÷ 1000. The diagram also shows the ampere-turns per inch length of path required for different densities in an air path. It will be noticed that there are two scales along the bottom of the chart; the smaller figures refer to the iron and steel curves and the larger figures to the straight line applying to air paths.

The use of the chart is simple. If it is desired to know the excitation required to get a given density, find that density in the vertical scale, trace horizontally into the chart until the line being followed reaches the curve of the material to be used; then trace straight down or up from that point to the horizontal scale, and the ampere-turns per inch of path length will be found. For example, sup-

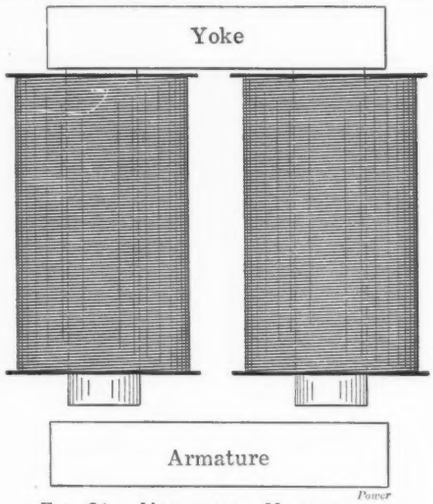


Fig. 21. Horseshoe Magnet and Armature

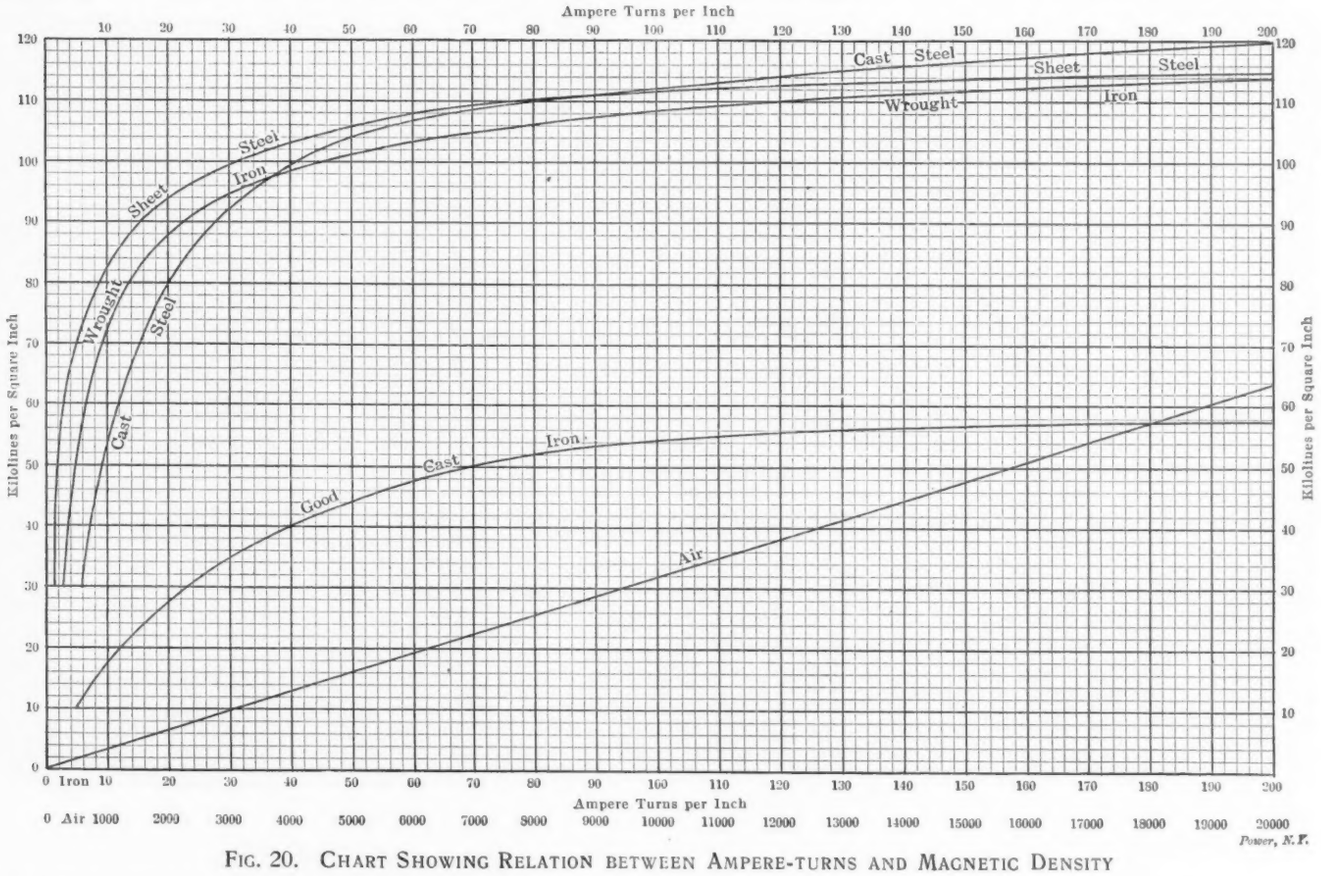


Fig. 20. Chart Showing Relation between Ampere-turns and Magnetic Density

pose you wished to produce a density of 92,000 lines per square inch in a wrought-iron core; starting at the point in the left-hand scale corresponding to 92, trace horizontally to the right until you strike the curve marked "Wrought iron," and then follow a vertical line from that point up to the scale at the top of the chart; here you will reach a point half-way between 24 and 26, and this, of course, would be marked 25 if the scale were divided into units and each one marked. This means that 25 ampere-turns must be provided for each inch of length in the core. In addition to this, excitation must be provided for whatever other parts of the magnetic circuit there may be, as indicated in the example already given.

In order to practise the application of the chart to imaginary or actual electromagnets other information is necessary; that is, what can be done with the magnetism produced in the magnet. The most common use of small magnets is that of moving light mechanism of some sort. If a piece of soft iron or steel be held in front of the poles of an electromagnet, as indicated in Fig. 21, and a current be passed through the winding of the magnet, the piece of iron will be pulled toward the magnet poles. The force of this pull will depend on the size of the magnet pole face, the quantity of flux passing from the pole faces to the piece of iron, and the distance between the pole faces and the iron strip. It is impracticable to find out accurately by calculation what the pull will be, but it can be estimated roughly.

If the iron strip were in actual contact with the pole faces, as in Fig. 22, and the quality of the iron in the magnet core and the armature (the piece of iron bridging the poles is called an "armature") were known, the pull could be calculated accurately. The pull, measured in pounds, when the armature is in contact with the pole faces, is equal to the area of the pole faces × the square of the magnetic density at the point of contact with the armature ÷ 72,134,000. Expressing this as a formula, and representing the density by B, the area of each pole face by A and the pull in pounds by F,

$$\frac{2AB^2}{72{,}134{,}000} = F.$$

For convenience in applying this formula and the chart of Fig. 20, the accompanying table has been prepared. This gives the pull that will be exerted by any electromagnet upon an armature in actual contact with its pole faces. Suppose, for example, the magnet shown in Fig. 21 were made of good soft wrought iron, the cores being ⅝ inch diameter and 3 inches long, and the yoke was 9/16 inch thick, 13/16 inch wide and 2 inches long from center to center of the magnet cores; also, that the armature was the same size as the yoke, and weighed ½ pound. What magnetic density would be required in the various parts of the circuit to enable the armature to hold up a weight of 16 pounds, and how much excitation would the winding and exciting current have to provide to maintain that density?

TABLE 4. MAGNETIC PULL AT DIFFERENT DENSITIES.

Magnetic Density per Square Inch at Point of Contact.	Pull in Pounds per Square Inch of Contact Area.	Pull, in Pounds per Pole, Exerted by Round Cores. Diameter of Poleface, Inch Measure.								
		½	9/16	⅝	11/16	¾	13/16	⅞	15/16	1
10,000	1.38	0.27	0.34	0.42	0.51	0.61	0.71	0.83	0.95	1.08
10,500	1.50	0.29	0.37	0.46	0.55	0.66	0.77	0.90	1.03	1.17
11,000	1.65	0.32	0.41	0.50	0.61	0.73	0.85	0.99	1.15	1.29
11,500	1.80	0.35	0.44	0.55	0.66	0.79	0.93	1.08	1.24	1.41
12,000	2.00	0.39	0.49	0.61	0.74	0.88	1.03	1.20	1.38	1.57
12,500	2.15	0.42	0.53	0.66	0.79	0.95	1.11	1.29	1.48	1.68
13,000	2.30	0.45	0.57	0.70	0.85	1.01	1.19	1.38	1.58	1.80
13,500	2.50	0.49	0.62	0.76	0.92	1.10	1.29	1.50	1.72	1.96
14,000	2.7	0.53	0.67	0.82	1	1.19	1.4	1.62	1.86	2.12
14,500	2.9	0.56	0.72	0.89	1.07	1.28	1.5	1.74	2.00	2.27
15,000	3.1	0.61	0.77	0.95	1.15	1.37	1.6	1.86	2.14	2.43
15,500	3.3	0.64	0.82	1.01	1.22	1.45	1.71	1.98	2.27	2.59
16,000	3.5	0.68	0.87	1.07	1.3	1.54	1.81	2.1	2.41	2.74
16,500	3.75	0.73	0.93	1.15	1.39	1.65	1.94	2.25	2.58	2.94
17,000	4	0.78	0.99	1.22	1.48	1.76	2.07	2.4	2.76	3.14
17,500	4.25	0.83	1.05	1.3	1.57	1.87	2.2	2.55	2.93	3.33
18,000	4.5	0.88	1.11	1.38	1.67	1.98	2.33	2.7	3.1	3.53
18,500	4.75	0.93	1.18	1.45	1.76	2.09	2.46	2.85	3.27	3.73
19,000	5	0.98	1.24	1.53	1.85	2.2	2.59	3	3.45	3.92
19,500	5.25	1.03	1.3	1.61	1.94	2.32	2.72	3.15	3.62	4.12
20,000	5.5	1.08	1.36	1.68	2.04	2.43	2.85	3.3	3.79	4.32
20,500	5.75	1.13	1.42	1.76	2.13	2.54	2.98	3.45	3.96	4.51
21,000	6	1.17	1.49	1.84	2.22	2.65	3.11	3.6	4.14	4.71
21,500	6.35	1.25	1.59	1.96	2.37	2.82	3.31	3.84	4.41	5.02
22,000	6.7	1.31	1.66	2.05	2.48	2.96	3.47	4.02	4.62	5.26
22,500	7	1.37	1.74	2.14	2.59	3.09	3.63	4.2	4.83	5.49
23,000	7.3	1.43	1.81	2.24	2.71	3.22	3.78	4.39	5.03	5.75
23,500	7.6	1.49	1.88	2.33	2.82	3.35	3.94	4.57	5.24	5.96
24,000	7.9	1.55	1.96	2.42	2.93	3.49	4.09	4.75	5.45	6.2
24,500	8.3	1.63	2.06	2.54	3.08	3.66	4.3	4.99	5.72	6.51
25,000	8.6	1.68	2.13	2.63	3.19	3.8	4.45	5.17	5.93	6.75
26,000	9.3	1.82	2.31	2.85	3.45	4.1	4.82	5.59	6.42	7.3
27,000	10	1.96	2.48	3.06	3.71	4.41	5.18	6.01	6.90	7.85
28,000	10.8	2.12	2.68	3.31	4	4.77	5.6	6.49	7.45	8.48
29,000	11.6	2.27	2.88	3.55	4.3	5.12	6.01	6.97	8	9.11
30,000	12.4	2.43	3.08	3.8	4.6	5.47	6.43	7.45	8.55	9.73
31,000	13.2	2.59	3.28	4.05	4.9	5.83	6.84	7.93	9.11	10.36
32,000	14	2.74	3.48	4.29	5.19	6.18	7.26	8.41	9.66	10.99
33,000	15	2.94	3.72	4.6	5.56	6.62	7.77	9.02	10.35	11.77
34,000	16	3.14	3.97	4.91	5.97	7.06	8.29	9.62	11.03	12.55
35,000	17	3.33	4.22	5.21	6.31	7.51	8.81	10.22	11.72	13.35
36,000	18	3.53	4.47	5.52	6.68	7.95	9.33	10.82	12.42	14.13
37,000	19	3.73	4.72	5.83	7.05	8.39	9.85	11.42	13.11	14.92
38,000	20	3.92	4.97	6.13	7.42	8.83	10.37	12.02	13.8	15.7
39,000	21	4.12	5.21	6.44	7.79	9.27	10.88	12.62	14.49	16.48
40,000	22	4.32	5.46	6.75	8.16	9.72	11.4	13.22	15.18	17.27
41,000	23	4.51	5.71	7.05	8.53	10.16	11.92	13.82	15.87	18.05
42,000	24.25	4.76	6.02	7.44	9	10.7	12.57	14.57	16.73	19.04
43,000	25.5	5	6.33	7.82	9.46	11.25	13.22	15.32	17.6	20
44,000	26.75	5.25	6.64	8.2	9.93	11.8	13.87	16.07	18.46	21
45,000	28	5.49	6.95	8.59	10.09	12.35	14.51	16.82	19.32	22
46,000	29.3	5.75	7.28	8.99	10.87	12.95	15.18	17.61	20.22	23
47,000	30.6	6	7.6	9.38	11.35	13.5	15.85	18.4	21.12	24.02
48,000	31.9	6.26	7.92	9.78	11.84	14.09	16.53	19.17	22	25.05
49,000	33.2	6.52	8.25	10.18	12.32	14.65	17.21	19.95	22.91	26.07
50,000	34.6	6.79	8.59	10.6	12.84	15.28	17.93	20.8	23.87	27.17
51,000	36.2	7.1	8.92	11.1	13.42	15.88	18.63	21.6	28.78	28.19
52,000	37.8	7.41	9.25	11.6	14	16.58	19.33	22.4	25.69	29.22

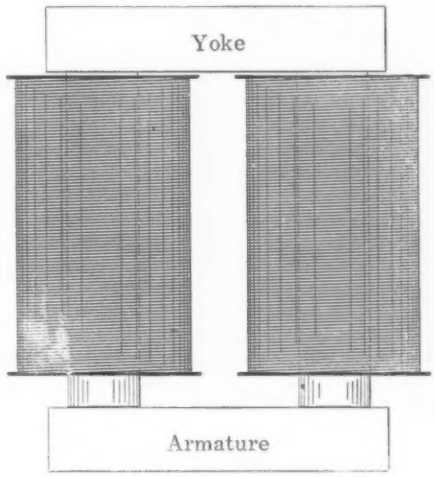


FIG. 22. ARMATURE IN CONTACT WITH POLES

Referring to Table 4 it will be found that a ⅝-inch round core will give a pull of 8.2 pounds per pole at a density of 44,000 lines per square inch at the pole faces; therefore, the two poles will pull 16.4 pounds at that density. But the armature weighs ½ pound, and its weight must be sustained in addition to the 16 pounds that it is to hold up; the total magnetic pull, therefore, must be 16½ pounds, or 8¼ pounds per pole. This is just beyond the pull that 44,000 lines density will give, and referring again to the table it will be found that at a den-

sity of 45,000 lines per square inch the magnet will hold 8.59 pounds per pole, which gives a sufficient margin to be sure of getting the 16 pounds of net holding power.

With the armature in contact with the poles, the leakage of magnetic flux will be so small that it is not worth while to consider it, especially in view of the fact that the density in the magnet cores and armature will be low. It will be practical, therefore, to consider that all the flux produced within the coils will pass through the yoke and armature. With a density of 45,000 at the pole faces, the density in the cores will be the same and the density in the yoke and armature will be practically

$$\frac{0.307}{0.46} \times 45{,}000 = 30{,}000$$

lines per square inch, the cross-sectional area of the yoke and the armature being 0.46 square inch while that of the core is 0.307 square inch. The various figures for excitation are more easily understood in tabular form, thus:

	Length of Path.	Density.	A.T. per Inch of Path.	A.T. for Path Length.
Cores	6	45,000	4	24
Yoke and armature	4	30,000	3	12
Total ampere-turns				36

To anyone accustomed to figure magnet windings this excitation of 36 ampere-turns will be absurdly small, because even very small magnets require usually several hundred ampere-turns of excitation. The explanation is that a magnet is ordinarily made to lift its armature from a distance, and the moment there is an airgap between the armature and the pole faces, no matter how small, the required ampere-turns go up to an astonishing degree. Suppose, for example, that the magnet had to lift the armature from a short distance, as indicated in Fig. 21, the airgap between the armature and the pole faces being ⅛ inch; then, even if we assume that there is no magnetic leakage, which could not be true, and assuming that the average area of each airgap was twice the area of the pole face, so that the density in the airgap would be 22,500 lines per square inch, the ampere-turns required for the airgaps would be 7050 per inch of length, and as the two gaps are ¼ inch in length, the ampere-turns required to carry the flux across them would be 1763. This would carry the total required ampere-turns up to 1799, or practically 1800, which would be divided equally between the two coils, putting 900 ampere-turns in each coil.

However, the lifting power of an electromagnet cannot be calculated accurately when the armature is not in contact with it, as already explained, because a large portion of the flux created in the cores leaks from the pole ends back to the yoke, and also across from pole to pole, without passing through the armature at all. The best that can be done with a magnet of this form is to assume a leakage of about 50 per cent. of the flux and take the density at the face of the armature as the criterion of pulling power. When the airgap is very small, as in the case of a telegraph sounder, or an ordinary electric bell, it is not so difficult to estimate the pull.

When it is desired to lift weights of considerable magnitude—anything over a pound or so—and especially if the load is to be lifted an appreciable distance,

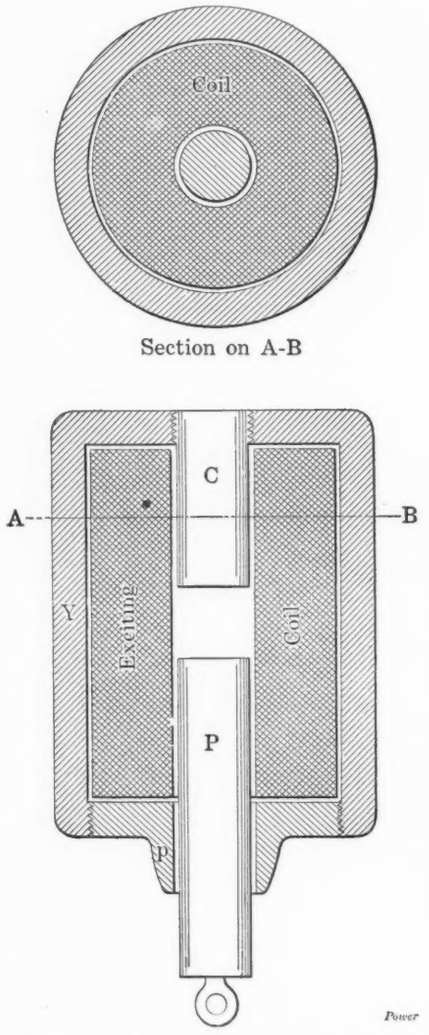


FIG. 23. PLUNGER TYPE OF ELECTROMAGNET

such as ¼ inch or more, magnets of the plunger type are employed. Fig. 23 is a sketch of this type in a form that is widely used. It consists of a short core *C* attached permanently to a yoke *Y*, frequently made of flat wrought iron, and movable core or plunger *P*, which fits as snugly as practicable in a hole through the lower part of the yoke. The dotted lines in Fig. 24 indicate the "direction" of the magnetic lines of force created by the exciting coil when the polarity of the fixed core *C* is "south;" the flux divides at the junction of the core and the yoke and goes through the yoke "in parallel" until it reaches the hole through which the plunger *P* slides; here the two divisions of the flux unite and

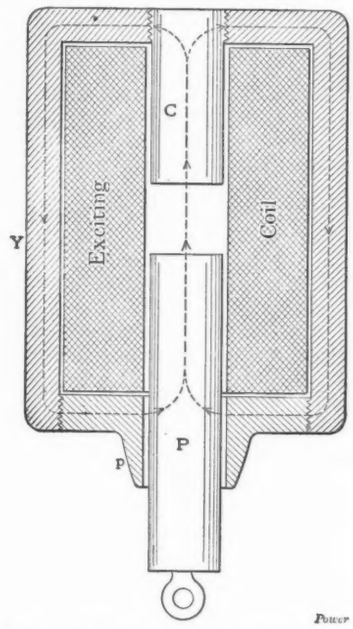


FIG. 24. DIRECTION OF MAGNETIC LINES OF FORCE

complete the circuit through the plunger and airgap. An extension *p* is provided around the hole where the plunger enters in order to increase the area of the airgap there and thereby reduce the magnetic density in that gap and, consequently, the ampere-turns required to force the flux across to the plunger. It makes no difference what polarity is given to the cores; the fixed core may be "north" and the plunger "south," or the reverse, as the case may be. With this sort of a magnet the magnet leakage is so small that its effect can be disregarded if a slight margin is allowed in excess of the calculated excitation. This is so because the exciting coil surrounds the airgap between the pole faces, where the greatest leakage would otherwise occur.

With a magnet of the plunger type the pull obtainable with a given excitation in ampere-turns is greater than with any other type, and it is practicable to get a much longer range of action than with the ordinary horseshoe type or any of its modifications, for the same reason, namely, that the inclosure of the airgap by the exciting coil reduces the leakage to the smallest possible proportion.

In 1906 there were 51,000 locomotives in service on the railroads of the country, which, it is estimated, consumed 90,000,000 tons of fuel at a cost of $170,500,000.

CORRESPONDENCE

A Puzzling Synchronizing Trouble

In the accompanying sketch is shown the plan of connecting the synchronizing circuits for three three-phase, 400-volt alternators, which recently gave rise to a peculiar trouble which resulted in a severe burn on the hands of the switchboard attendant. As will be seen, both lamp and synchroscope methods are used.

For the synchroscope a synchronizing bus runs to each of the generator panels, leads being run from the bus to the eight-point receptacles *A*, *B* and *C*. When the fused plug is inserted connection is made between the top and bottom synchronizing bus (which is also a voltmeter bus) directly to the generator leads.

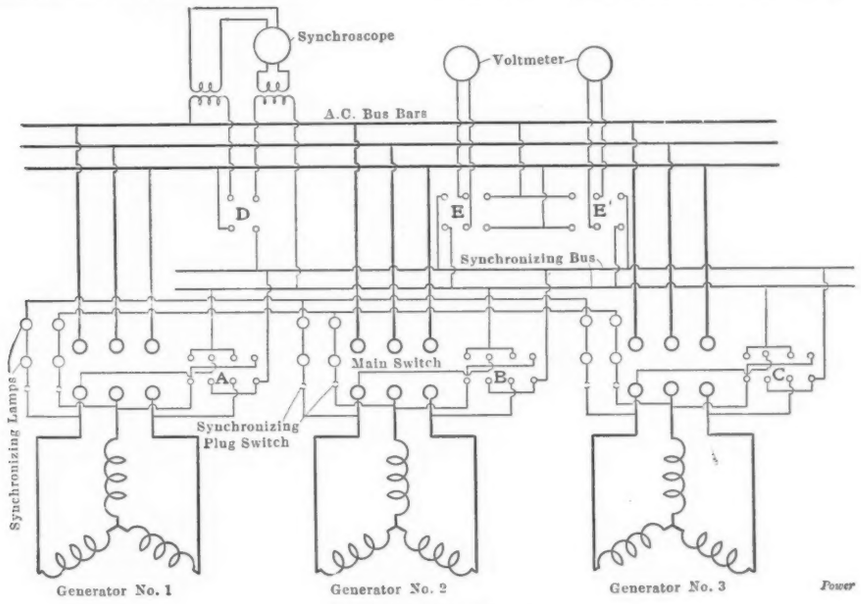


DIAGRAM OF WIRING

To use the synchroscope a plug is inserted in the middle position of *A*, *B* or *C*, depending on which generator is to be brought in, and at *D*, which cuts the two synchronizing transformers into service. To use the lamps, the plugs are inserted on two panels, one of which must be in service, and the lamps indicate synchronism of *the two armatures* instead of armature and main bus bars, as is usual in the direct-lamp method.

In synchronizing it is the practice to use the synchroscope, using the lamps only as a check on the accuracy of the former. In the case mentioned it was noticed one evening that when the plug was inserted at *C* or *A* the lamps would come up to full candlepower and flicker in a most peculiar manner. It was also noted that one set would burn in this manner with the plug switches on, the other set open. If the plug was removed from *A* or *C*, the lamps would behave in their normal manner. It is, of course, understood that plugs were not inserted at *A* and *C* at the same time, but were put in each place as its respective generator was being brought up.

A portable bank of lamps was hung across the main switch, their action being the same as the regular lamps. Seeing the lamps were worthless as a check on the synchroscope it was decided to risk the "scope," which seemed to behave in its usual manner.

In this case generator No. 3 was being brought up, the plug being inserted at *C*. When synchronism was indicated the main switch was closed and at the same instant the plug *C* blew up causing a short-circuit across the receptacles, the resultant flash of which severely burned the hands of the operator, the plug and the plug receptacles.

That the generator was in synchronism is proven by the fact that there was not the slightest disturbance in the system, the recording voltmeter on the direct-current side of the rotaries not showing any unusual variation. The three rotaries are operated directly off the alternating-current busbars.

Some loose connections at the receptacles were found, which, together with dust on the back of the board, was advanced as a reason for the flash. The board was carefully dusted and the dust collected in a box and a test made as to its conductivity. No deflection of the voltmeter could be secured through the dust at 1/16-inch distance. The potential receptacles are about an inch apart and surrounded by an insulating bushing, so this seems to preclude the dust theory.

And if the cause was dust, would not the flash have occurred across the back of the board instead of the front? Could loose contacts have caused the trouble? Would not such rather increase the resistance, which would tend to prevent the current jumping across between the receptacles? The actual cause seems to have never been discovered, although a possible cause may be found in the surmise that some of the wiring was disarranged by workmen who had been working around the board, although nothing was found to indicate this. After new receptacles were put in conditions were normal and have remained so.

C. L. GREER.

Handley, Texas.

Electric Drive for Factories

W. J. MacIntire's article in the November 16 number on "Electric Drives for Factories" brings to mind a change that was made from shafting and belt to motor drives. With the engine, shafting and belts, and it was a well laid out plant, we used 400 horsepower—we used only 300 horsepower with the motor drive.

There were 30 motors used in the plant. All machines that required over five horsepower were equipped with individual motors, and the machines using less than five horsepower were grouped together and driven from a line shaft which was driven by a large motor.

All of these motors were of the induction type and with the results obtained I would not think of using any other but the induction motor for factory work.

Mr. MacIntire says that with the induction motors the power factor was low—from 0.65 to 0.75—for which I can see no good reason, as in the above-mentioned plant the power factor was from 0.85 to 0.95, and this power factor can be obtained in any well designed plant.

The motors gave us no trouble at all, although sometimes through carelessness on the part of the motor tender, he would burn out a bearing, which is liable to happen on any machine.

Some of these motors were in very dusty and dirty places, and I know that no direct-current motor could stand the overload, wear and tear and abuse that the induction stood in this plant.

H. H. BERG.

Orange, N. J.

GAS · POWER DEPARTMENT ·

Everything worth while in the Gas Engine and Producer Industry will be treated here for the use of practical men

A Simple Oil Gas Producer

A gas producer that is said to make tar-free gas from crude oil was described in a paper read recently by A. B. Davis before the Ohio Society of Mechanical, Electrical and Steam Engineers, and some results obtained with the producer were presented and discussed. Fig. 1 is a sectional view of the nozzle used in this producer, which is known as the Darling. There are two jets delivering into the nozzle proper; oil is fed through the upper jet and regulated by means of a needle valve; a sight-feed glass is provided at *S* through which to note the rate of feeding. Air is fed in at the lower jet *A*, and the supply is controlled by means of the valve at the left. The air orifice is ⅛ inch in diameter, and the orifice in the end of the nozzle, through which the mixed air and oil pass to the furnace, is 3/16 inch in diameter. The nozzle delivers the mixture into a retort provided with a system of baffles which serve to improve the intimacy of mixture and to retard the flow so as to obtain the desired temperature for the production of gas. Fig. 2 shows one of these retorts with the nozzle attached.

There are two groups of retorts, one above the other. The lower group is heated by means of a burner to which oil and air are fed by a nozzle exactly like those which feed into the retorts. This group of retorts delivers into the upper group, which is not heated. Fig. 3 shows a 32-retort producer equipped with two burners, one under each lower group of eight retorts. The upper retorts deliver the gas into a manifold whence it is taken to a holder.

Concerning the absence of tar in the gas produced, Mr. Davis says:

The remarkable thing about the producer in question is that it can be operated without the production of tar, while all other processes known to the author, producing gas by the decomposition of oil at a high temperature, also produce more or less tar, some processes converting into tar as high as 25 per cent. of the total volume of oil fed to the retorts. Carbureted water-gas machines convert into tar on an average about 20 per cent. of the oil used in carbureting the gas. Several processes on the market, similar to the one under consideration, "crack up" the oil in a hot retort in the presence of steam and produce large amounts of tar.

The question arises: Why is it possible to "crack up" the oil in the presence of air at a high temperature without producing tar, while, if steam or inert gases be used, tar is produced? We well know that oil alone in the retort will produce

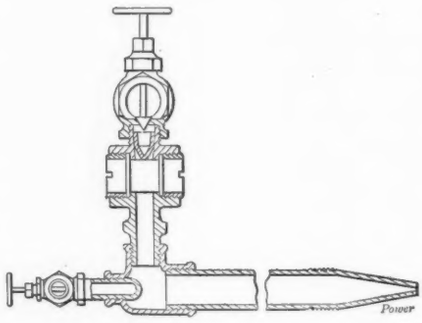


FIG. 1. MIXER AND NOZZLE OF DARLING PRODUCER

tar, just as in the Pintsch process used to make gas for railway cars. This gas may be easily made with the apparatus described by shutting off the flow of air into the retorts, and then, as in the Pintsch process, about 20 per cent. of the oil is converted into tar. After a careful study of the process, Mr. Davis came to the conclusion that the explanation of the lack of tar in the gas was as follows:

The composition of crude oil may be represented by the formula

$$C_nH_{2n+2}, \text{ or } H\text{—}\underset{H}{\overset{H}{C}}\text{—}\underset{H}{\overset{H}{C}}\text{—}\underset{H}{\overset{H}{C}}\text{—}\underset{H}{\overset{H}{C}}\text{—}C_nH_{2n+1}.$$

The oil when heated to a high temperature shows a tendency to crack up into hydrocarbons of lower molecular weight, at the same time producing compounds of the olefine type from the paraffins by causing the oil to drop two hydrogen atoms at one or several points in the chain. These olefine compounds may be represented by the formula

$$C_nH_{2n},$$

or

$$CH_2 = CH\text{—}C_nH_{2n+1},$$

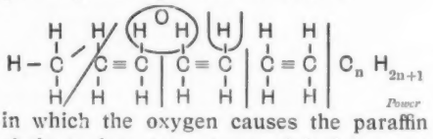
or

$$CH_2 = CH\text{—}CH_2\text{—}CH_2 \quad C_nH_{2n+1}.$$

The formation of paraffin oil tar depends on the polymerization of these high olefines which always takes place when they are heated to a high temperature. The tar may be represented by the general formula

$$(C_nH_{2n+1}\text{—}CH = CH_2)x.$$

It may be considered a mixture of many compounds, some of which are of very high molecular weights, which have resulted from the combination of the olefines—and the compounds forming the tar may be considered to have one or several benzene rings in the molecule and with one or many side chains on each ring, perhaps intricately connected among themselves. It has been found that the decomposition of the oil into gas takes place best at a temperature of 1350 to 1470 degrees Fahrenheit. The reactions which take place in the presence of oxygen may be expressed as below:



in which the oxygen causes the paraffin chain to lose two atoms of hydrogen at several points in the chain, causing double linkages, which are always weak spots and which allow the chain to be broken up by the heat into ethylene, acetylene and hydrogen, principally. The presence of oxygen in the retort seems to cause the olefines formed to break down at once into permanent gases without allowing them to polymerize to tar. As proof that such a reaction takes place, the analysis of the gas shows that it contains 25 to 30 per cent. of gases soluble in bromine water, these being largely ethylene and acetylene. Furthermore, the gas contains very little carbon dioxide and carbon monoxide, showing that the oxygen of the air had combined with the hydrogen of the oil to form water, rather than combining with the carbon to form either carbon dioxide or carbon monoxide. In order to settle more definitely to the author's satisfaction the merits of the conclusions that the oxygen from the

air used prevented the olefines from the oil from polymerizing, the problem was presented to Dr. J. Bishop Tingle during a meeting of the American Chemical Society in Detroit this summer, who, after some study, expressed the opinion that the above conclusion in this matter is probably correct. The problem was also placed before Dr. William A. Noyes, of the Bureau of Standards, who answered the author's letter as follows:

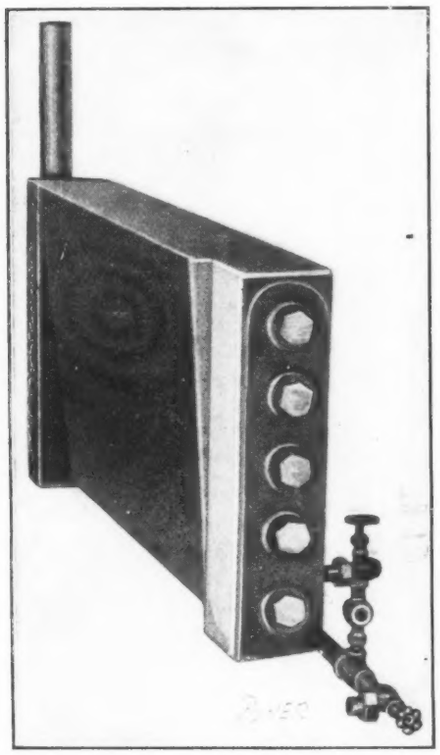


FIG. 2. RETORT WITH NOZZLE ATTACHED

"The explanation which you give for the lack of tar in your process is probably partly correct. I think also that in the original decomposition by heating, the free valences of the carbon, which under ordinary circumstances caused the polymerization, are saturated by oxygen or hydroxyl followed by the loss of water in such a manner that the resulting hydrocarbons do not readily polymerize. Just what happens is hard to determine, and I do not, for the moment, see any very direct method of experimental study of the question. The fact is, of course, very important technically. From the scientific point of view it is possible that there are a great many different factors which unite to give the results obtained."

A test made of this producer recently with Lima "gas" oil gave the following results:

Oil delivered to retorts per hour	19 gal.
Oil recovered from scrubber per hour	8.5 gal.
Oil converted into gas per hour	10.5 gal.
Oil used in burner per hour	4 gal.
Total oil actually used per hour	14.5 gal.
Quantity of gas made per hour	1486 cu.ft.
Oil used in retorts per 1000 feet of gas	7.07 gal.
Total oil used per 1000 feet of gas made	9.76 gal.
Heat value of oil per gallon	133,000 B.t.u.
Heat value of gas per cubic foot	844.5 B.t.u.
Heat efficiency of the retorts alone = 844.5 ÷ 939.8 × 100 = 89.8 per cent.	
Heat efficiency of complete producer = 844.5 ÷ 1298 × 100 = 65.1 per cent.	
Air used per hour, atmospheric-pressure basis	583 cu.ft.
Pressure of air delivered to nozzles per square inch	35 lb.
Air per 100 feet of gas, atmospheric-pressure basis	392.3 cu.ft.

The chemical analysis of the gas produced was as follows:

	Per Cent.
Carbon dioxide	2.0
Illuminants soluble in bromine	25.5
Heavy hydrocarbons	0.5
Carbon monoxide	1.5
Methane	35
Hydrogen	5.5
Nitrogen	30

In April of last year the author spent a week at an experimental plant in Cleveland to develop the gas for lighting purposes as well as power. When a very low grade of gas was produced, or even gas up to 600 or 700 B.t.u. per cubic foot, no tar was produced; but with very high candlepower gas considerable tar was obtained. On all gas engines the gas works perfectly, and in all types of gas lights made for water or coal gas there is no difficulty in adjusting the lights so that the producer gas gives most excellent results. There has been some trouble with gas stoves owing to the fact that the specific gravity of the producer gas is just about the same as air; in some cases slightly higher. The specific gravity of coal gas is only about 0.48, and carbureted water gas 0.65, natural gas 0.55 to 0.6, so that practically all the stoves on the market are made to use a gas which is about one-half as heavy as the oil gas made by this producer.

According to Bunsen's law, the rates of flow of two gases through an orifice are to each other as the square roots of their gravities. Therefore, since this gas is twice as heavy, it will flow as 1 to 1.414, when compared with gas of a gravity of 0.5. For this reason and because of the large quantity of nitrogen present from the air, only the oxygen of which has been used, the small and numerous orifices in a gas-stove burner are not, as a rule, large enough to admit the oil gas flowing through rapidly enough and have to be drilled out about 50 per cent. larger. Unless this is done, the flame of a rose burner has a marked tendency to blow itself out, as any gas will do when fed to a burner under too high pressure. This difficulty is most easily overcome by ordering burners directly from the factory, these being drilled for the producer gas. In all other appliances on which the producer gas has been tried, it either works well without change of adjustment, or else the appliance can be adjusted without change in construction.

From an engineering standpoint, no doubt, many improvements can be made in the design of the setting of the apparatus as it stands at present. This is usually the case with any apparatus when first gotten out. For instance, a steam boiler might well be placed on top of the setting and the waste heat from the retorts would, no doubt, furnish more than enough power to compress the air used. Furthermore, preheating the air by waste heat, and proper asbestos or mineral-wool lining in the setting, would all make it much more economical to operate than at present.

Since the gas is made entirely from oil it need not be purified, provided the sulphur contents of the oil are as low as one-tenth of one per cent.

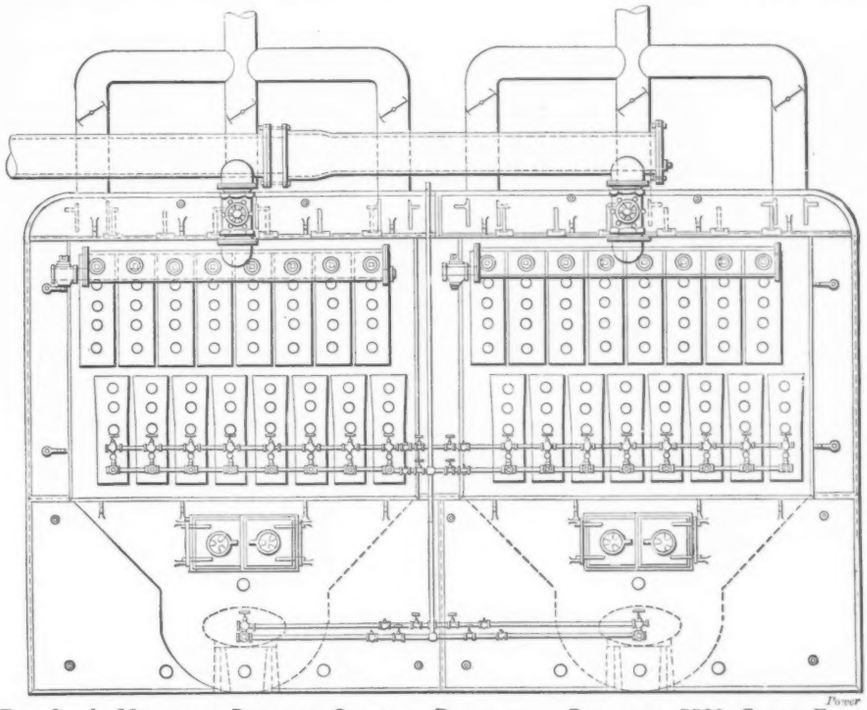


FIG. 3. A 32-RETORT DARLING OIL-GAS PRODUCER. CAPACITY, 5500 CUBIC FEET AN HOUR

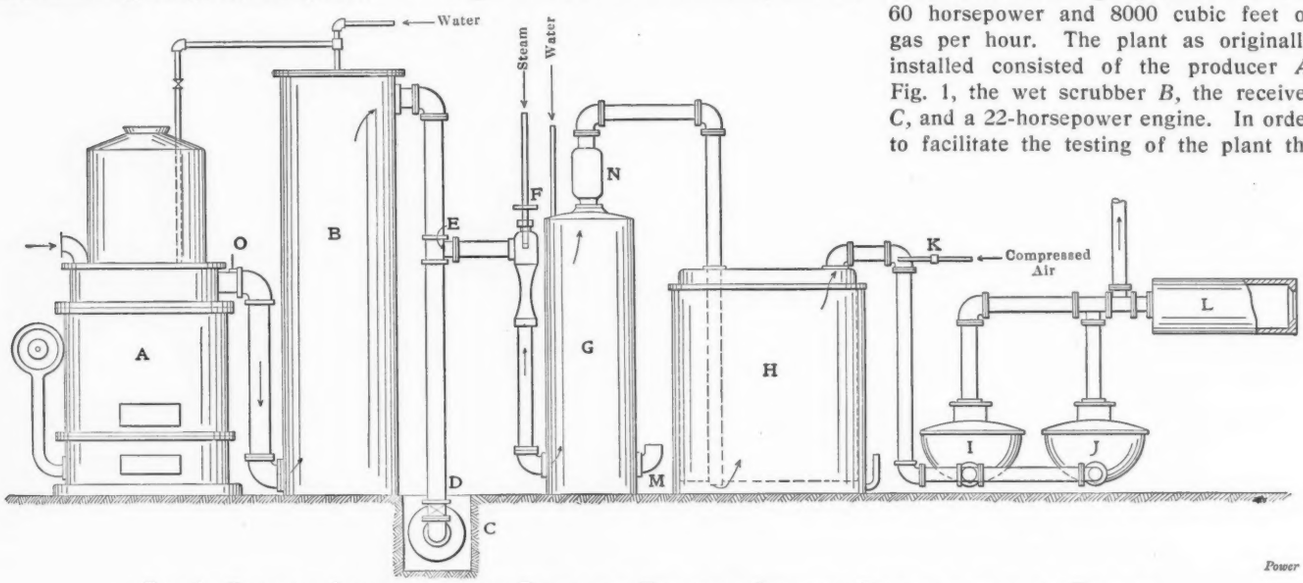
Test of an Otto Suction Gas Producer*

By C. M. Garland and A. P. Kratz

The method of testing the suction gas producer herein described has been used by the writers to advantage in tests made in the mechanical-engineering laboratory of the University of Illinois. The method of testing has reduced the labor of running such tests to a minimum, and the forms for computation have greatly reduced the labor and tedium of the calculations.

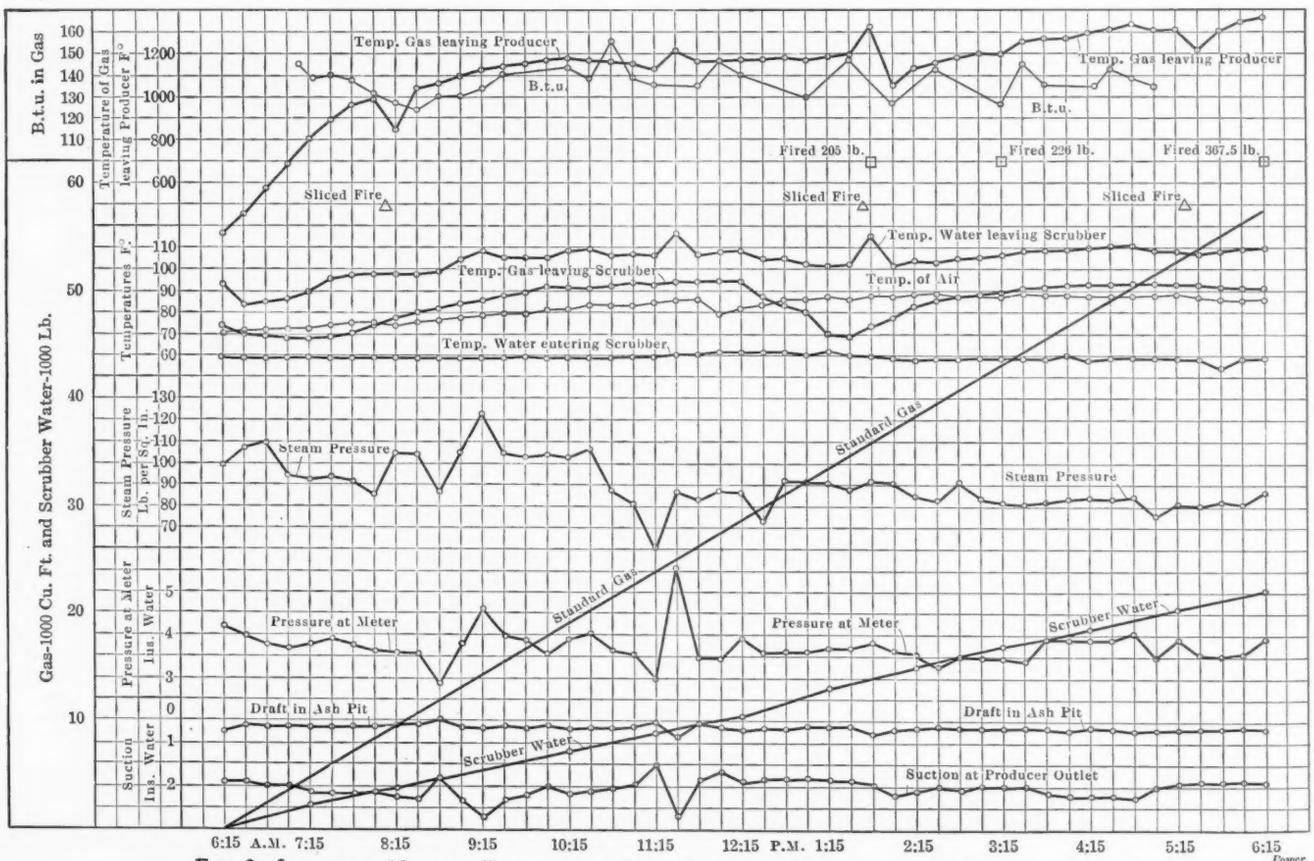
The tests at the university were made on an Otto suction gas producer rated at 60 horsepower and 8000 cubic feet of gas per hour. The plant as originally installed consisted of the producer *A*, Fig. 1, the wet scrubber *B*, the receiver *C*, and a 22-horsepower engine. In order to facilitate the testing of the plant the

*Abstract of a paper read before the Gas Power Section of the American Society of Mechanical Engineers, December 9, 1909.



Power

FIG. 1. GENERAL ARRANGEMENT OF PLANT FOR TESTING A SUCTION PRODUCER WITH AN EJECTOR


B.t.u. in Gas
Temperature of Gas leaving Producer F.°
Temperatures F.°
Steam Pressure Lb. per Sq. In.
Pressure at Meter Ins. Water
Suction Ins. Water
Gas-1000 Cu. Ft. and Scrubber Water-1000 Lb.
Temp. Gas leaving Producer
B.t.u.
Fired 205 lb.
Fired 226 lb.
Fired 367.5 lb.
Sliced Fire
Temp. Water leaving Scrubber
Temp. Gas leaving Scrubber
Temp. of Air
Temp. Water entering Scrubber
Standard Gas
Steam Pressure
Pressure at Meter
Scrubber Water
Draft in Ash Pit
Suction at Producer Outlet
6:15 A.M. 7:15 8:15 9:15 10:15 11:15 12:15 P.M. 1:15 2:15 3:15 4:15 5:15 6:15

Power

FIG. 3. LOG OF A 12-HOUR TEST OF AN OTTO PRODUCER MADE WITH A KOERTING EJECTOR

connection to the engine was blanked and a Schutte-Koerting steam ejector of 12,000 cubic feet hourly capacity, inserted in the gas main at *F*, was used to draw the gases from the producer and deliver them to the wet scrubber *G*, where the steam used by the ejector was condensed.

The condensed steam and condensing water pass out at the overflow *M*, while the gases pass out through the separator *N* and into the drier *H*, constructed from a gas bell or holder, and filled with straw, and used to separate the suspended moisture from the gases before they enter the meters *I* and *J*. The meters used in our tests were of 8000 and 3500 cubic feet hourly capacity, respectively, and were connected in parallel for capacities greater than 8000 cubic feet per hour, the larger meter alone being used for lower capacities. From the meters the gases were discharged into the atmosphere above the roof of the laboratory.

A gage box *L*, adapted to receive thin plates with orifices, is used in calibrating the meters, by means of air. The meters having been blanked from the gas main, compressed air is admitted at *K*, and expanding passes through the meter to be calibrated and out at the orifice in *L*. The data for the orifice were taken from the paper by R. J. Durley, in Volume 27, 1906, of the *Transactions*. After the calibration, the inlet to the box was blanked off.

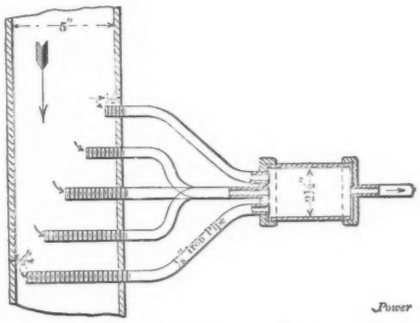


FIG. 2. APPARATUS FOR TAKING GAS SAMPLES CONTINUOUSLY

The gas generator is of the contained-vaporizer type, with grate and without charging bell, the specifications stating that it is only to be used 12 hours at a time. During some of the earlier tests the cast-iron vaporizer cracked. A steam jet was then used to supply the moisture, and the vaporizer was blanked off. The weight of steam was measured by passing the jet through a calibrated orifice in a thin plate.

Carrying out a test with the ejector instead of an engine does not differ essentially from the procedure when an engine is used, so far as the producer is concerned. The test started with the producer full and with a clean fuel bed. The coal fired during the test is weighed and at the end of the test the fire is cleaned, the fuel bed being brought to as near the starting condition as possible, and the producer filled. In order that the error in determining the weight of coal fired in this manner might be known, the producer when cold is filled a number of times, and the weight of coal required is noted. The average of these weights is taken to be the true weight of coal required to fill the generator, the probable error in filling with a given weight of coal being estimated from these results. It is endeavored to make the tests of such duration as to bring the probable error of filling down to about 2 or 3 per cent.

The temperature of the gas leaving the generator is taken at *O* by means of a platinum-rhodium thermo-couple and a Siemens & Halske milivoltmeter, calibrated to read direct in Centigrade degrees. The temperatures in the fuel bed are taken with Hoskins thermo-couples and galvanometer, the latter reading in degrees Fahrenheit. Other temperatures are taken with mercury thermometers.

In a recent test the temperatures in the fuel bed were taken in three horizontal zones 10 inches, 18 inches and 24 inches, respectively, above the grate. In each zone readings were taken 3 inches from the lining on each side, and in the center of the fuel bed. The results are given in Table 1. By means of the sampling tube illustrated in Fig. 2, samples of gas were taken continuously for test by a Junker calorimeter and for analysis, by Hempel apparatus. The results of the analyses are given in Table 3.

As already stated, the weight of steam fed to the producer is determined by the use of a calibrated orifice. By means of a small laboratory aspirator, a sample of the gas leaving the producer is drawn successively through a calcium-chloride

TABLE 1. TEMPERATURES IN FUEL BED.

Time.	Zone No.	Temp. 3 In. from Near Wall, Deg. F.	Temp. at Center, Deg. F.	Temp. 3 In. from Far Wall, Deg. F.
10:05–10:10 a.m.	1	2100	2037	2025
10:25–10:30 a.m.	2	2350	2225	2275
10:43–10:55 a.m.	3		2200	2400

TABLE 2. FUEL AND WATER DATA.

FUEL PER HOUR.

Dry coal fired per hour	64.7	lb.
Combustible consumed per hour	51.2	lb.
Dry coal per sq.ft. of grate area per hour	38.8	lb.
Combustible per sq.ft. of grate area per hour	30.7	lb.
Dry coal per sq.ft. of fuel bed per hour	34.5	lb.
Combustible per sq.ft. of fuel per hour	27.3	lb.
Rate of descent of dry coal through fuel bed per ft. per sq.ft. per hour	15.6	lb.
Rate of descent of combustible through fuel bed per ft. per sq.ft. per hour	12.4	lb.
Ash and refuse taken out per hour	7.1	lb.
Calorific value by oxygen calorimeter per pound dry coal	13,040	B.t.u.
Calorific value by oxygen calorimeter per pound of combustible	15,700	B.t.u.

WATER.

Total weight of steam fed to vaporizer		267.8	lb.
Total weight of water fed to producer		341.5	lb.
a Steam to vaporizer	267.8 lb.		
b In air	51.7 lb.		
c In coal	22.0 lb.		
Total weight of water decomposed		218.2	lb.
Total weight of water in gas leaving producer		123.3	lb.
Ratio of water decomposed to water supplied		0.639	lb.
Weight of water decomposed per pound gas generated		0.0558	lb.
Weight of water decomposed per pound of dry coal fired		0.281	lb.
Weight of water decomposed per pound of combustible consumed		0.355	lb.
Weight of water decomposed per pound of air supplied		0.0702	lb.
Weight of water supplied per pound of dry coal fired		0.440	lb.
Weight of water supplied per pound of combustible consumed		0.556	lb.
Weight of water supplied per pound of dry air used		0.1097	lb.
Total weight of scrubber water		22,200	lb.

TABLE 3. GAS ANALYSIS BY VOLUME.

Sample No.	Time Taken.	Per Cent. CO_2	Per Cent. O_2	Per Cent. CO	Per Cent. CH_4	Per Cent. H_2	Per Cent. N_2	B.t.u.
1	6:23– 9:05 a.m.	5.7	0.5	22.9	2.1	12.2	56.6	134
2	9:10–10:37 a.m.	4.1	0.2	27.9	1.6	11.1	55.1	142
3	10:39–12:25 p.m.	3.3	0.1	28.4	1.5	9.5	57.2	137
4	12:30– 2:52 p.m.	4.3	0.2	26.9	1.8	10.6	56.2	139
5	3:00– 4:35 p.m.	3.6	0.1	28.6	1.8	9.0	56.9	139
6	4:40– 6:05 p.m.	4.1	0.3	27.4	1.8	10.0	56.4	138
Average.		4.20	0.23	27.01	1.77	10.40	56.40	138.1

TABLE 4. HEAT BALANCE.

DEBIT	B.t.u.	
Total heat supplied per pound dry coal	13,040	
Total heat supplied by air per pound dry coal	19	
Total heat supplied in steam per pound dry coal	385	
Total	13,444	
CREDIT	**B.t.u.**	**Per Cent.**
Heat contained as sensible heat in dry gas	1,725	12.8
Heat contained in moisture	262	2.0
Heat contained in dry gas (heat of combustion)	10,240	76.2
Heat in unburned carbon	618	4.6
Heat lost in radiation and conduction	599	4.4
Total	13,444	100.0

tube and a small gas meter, the weight of moisture being determined by the calcium-chloride tube and the volume by the meter. The per cent. of moisture determined by this method is used merely as a check, the percentage used in the computations being obtained by calculating the weight of water decomposed from the analysis of the gases and the analysis of the fuel. The difference between this quantity and the total weight of moisture carried into the producer, gives the weight of the moisture in the gas leaving the producer.

The volume of gas generated by the producer, and measured by the meters, is also checked by computing the volume of the gas generated from the analyses of gas and coal. In the anthracite producer, where the loss of carbon in soot and tar is small, probably not over 1 per cent., this offers an excellent means of checking the gas volume and also of computing the weight of air used. The gas analysis, where continuous samples are taken by the form of sampling tube illustrated, should be accurate within 1 per cent. The greatest error is likely to be made in the sampling of the coal. With a fine coal, such as pea or buckwheat, and a sample representing from 10 to 20 per cent. of the total weight of coal fired, the error in sampling should not exceed 2 per cent. The maximum error in determining the gas volume and the weight of air used should not exceed 5 per cent., if the error in filling the producer is 2 per cent. The probable error is, therefore, much less. In most of our tests the volume of gas computed from analysis has checked within 5 per cent. the volume determined by the meters. The meters are known to be accurate well within 2 per cent.

In testing large producers for bituminous coal, it is often difficult to measure the gas volume by any mechanical means. In such cases, if the carbon lost in the soot and tar is estimated from a sample of the soot and tar, and this amount deducted from the total weight of carbon in the coal, the volume may then be computed from the analyses of the gas and coal, and may be relied upon within 5 per cent., provided the sampling is accurate.

The producer referred to has a grate 15x16 inches, giving an area of 1.667 square feet. The mean diameter of the fuel bed was 18½ inches and the depth 26½ inches; the area of the fuel bed being 1.88 square feet. The outside diameter of the generator shell is 2 feet 10 inches and the hight 7 feet 1½ inches.

The fuel used in the test under consideration was Scranton anthracite pea of 13,040 B.t.u. per pound of dry coal, as measured by the calorimeter, and 13,125 by analysis. The analyses were as follows:

Proximate.		Ultimate.	
Fixed C.....	78.45	C.....	79.84
Volatiles.....	5.99	H.....	2.67
Moisture.....	2.75	O.....	2.37
Ash.....	12.81	N.....	0.82
	100.00	S.....	1.13
Sulphur.....	1.10	Ash.....	13.17
			100.00

Table 2 shows the fuel and water supplied during the test, which lasted 12 hours. Table 3 gives a gas-analysis log, and Table 4 gives the heat balance.

It will be noted from Table 2 that the weight of ash and refuse was much less than the weight of ash alone that would be obtained by computing from the analysis. This was due to the difficulty in cleaning the ash out of the fuel bed, and partly to the loss of ash in the form of dust, which is carried over into the scrubber. In this particular coal, which had very little tendency to clinker, the ash was so soft and fine that it packed in and filled the interstices between the live coals. A small amount of clinker was formed on the sides. This tendency of the ash to pack in the fuel bed, while it prevents the accurate determination of the actual weight of ash, does not, it is believed, materially affect the determination of the weight of coal as fired, for the reason just given; that is, while the fuel bed may contain as much carbon at the start as at the close, the bed is much more compact, due to the ash. The weight of ash and refuse is valuable principally for the determination of the unburned carbon lost through the grate.

The weight of dry coal per square foot of grate area per hour was high; while the producer was operating only at about 4800 cubic feet per hour capacity, this was considerably above its actual capacity. If the fuel had contained a fusible ash the results as shown in the table and the graphical log, Fig. 3, would have been practically impossible.

Discussion

The discussion of this paper was rather meager. Prof. R. H. Fernald commended the method of testing a producer with an ejector, but questioned the accuracy of calibrated orifices as a means of measuring steam flow, unless the pressure of the steam and its quality could be kept absolutely constant throughout a test. His experience in the Geological Survey trials showed that there is a need of careful and systematic investigation of the proportion of steam required by various fuels. In six tests made at the Norfolk station, where the steam was supplied by a separate boiler and the quantity of feed water accurately measured, the steam consumption per pound of coal fired ranged from 0.69 to 1.12 pounds, although the composition of the coal was substantially the same in all six tests. He also pointed out that a large error may be incurred by running a test for a certain limited period and trying to bring the fuel bed to its original condition; if the fuel bed at the close of the selected period has not been brought by natural operation to the same condition as in the beginning of the test, the test should be continued until that condition is naturally attained. Forcing the condition of the fuel bed is productive of error because of the difficulty of determining accurately the depth of the bed and because, for a short run, the quantity of fuel normally contained in the generator is large compared with the quantity consumed during the test. He questioned whether the fuel bed in the producer tested by Mr. Garland was in the same condition at the end as at the beginning of the test, because of the shortness of the test period and the arbitrary limitation of it to the selected number of hours.

H. F. Smith said that the producer log presented in the paper showed that the condition of the fuel bed at the end of the test could not have been the same as at the beginning because the temperature of the gas was so much higher at the close.

Limitations to Rope Drives

Where large amounts of power are to be transmitted, the selection of rope drive is only limited by the allowable distance between shafting journals. A number of drives of 4000 horsepower are running with the best of satisfaction, while systems of from 1000 to 2000 horsepower are numerous. It is claimed that in drives of such magnitude, rope has a considerable advantage over leather belting, not only in first cost, but in safety. In the matter of first cost no comparison can be made above 1000 horsepower, and where centers of driving and driven shafts are nearer together than 25 feet or further apart than 35 feet, the rope is decidedly superior to leather with smaller powers. The first cost of leather in a belt drive, assuming the most favorable conditions for it, would be about 2½ times the cost of manila rope to transmit equal power. The cost of the pulleys in a rope drive would be about one-third more than the leather-belt pulleys, which, in a drive of large capacity, would leave a good margin in favor of rope. This margin increases rapidly in favor of the rope as the distances apart of shafts or amount of power transmitted increases. The matter of first cost may be carefully estimated in each particular drive and the comparative sum closely ascertained. --*Mining Science.*

Gasolene engines of the usual four-stroke-cycle stationary type will ordinarily consume about a pint of gasolene per brake horsepower-hour when operated at about rated load and with a reasonably favorable adjustment of the mixture, quality and time of ignition.

CORRESPONDENCE

Adjusting the Gas Engine

On one of my gas-engine erecting trips I was sent to a small town to set up a 2½-horsepower gas engine for a farmer. This man had a natural-gas well on his farm which would fill up with water and then the gas pressure would drop very low. He wished to pump out the well at regular intervals, and also use the engine power for grinding grist, sawing wood and running other machinery.

When I arrived the man who had driven the well was engaged in putting in a pumping rig. He claimed to be an all-round mechanic, and to know all about gas engines. He had an ordinary belted deep-well pump installed in good shape, and as the engine ran at 350 revolutions per minute a countershaft was required. He had figured originally on 300 revolutions per minute and only found out the difference after I arrived.

He had found a large wood pulley in a scrap pile, and, as this proved too small had fastened on wooden lagging about 2½ inches thick. Although the wheel was about 5 feet in diameter no attempt was made to balance it. Two timbers were set in the ground and beveled off. Cast-iron boxes were used. The well driller had difficulty in putting in lag screws, because he did not take the time to drill the hole first with brace and bit. So he got some large spikes and drove them in with the largest hammer procurable. Both boxes were cracked, but that seemed to make little difference to him.

All was ready at last, and I had a little trouble in starting, because the pressure regulator was hard to adjust. Natural gas is very rich, and on starting if the pressure is high the gas may rush in so fast as to get such a rich mixture that it will not explode. Finally I started the engine, and such a wabble of wheel and swinging of belts. In about five minutes the wheel decided to see what was going on at the well. Fortunately no innocent bystander was injured and the derrick was strong enough to stop the wheel.

Much good(?) advice was offered by the numerous bystanders, and after considerable discussion the repairs were decided upon.

On my arrival the next morning I found the well man gone and his rig repaired. The bottom half of both boxes had broken and so the timber was cut out to form the lower half. This time he had gotten in some lag screws to fasten down the upper half.

On looking over the engine I found that he had taken the screwdriver and wrench and changed every adjustment possible, also the weights in the pressure regulator, retarded the ignition, opened the gas valve to its widest limits, etc.

I never heard how the farmer finally overcame his pump-rig difficulty, but I do know that he ran the engine for a number of years afterward, and from what he said before I left I do not believe that he attempted to make many adjustments, as the other man's experience was enough for him.

LOUIS J. BUSCHMAN.

Cleveland, Ohio.

Pre-Ignition Trouble

During a recent test of a gas engine I was badly troubled with pre-ignition, as will be seen by the diagram, Fig. 1. The mean pressure was about 75 pounds per square inch, the engine giving about 190 brake horsepower.

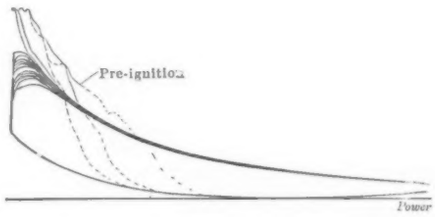


FIG. 1. SHOWING PRE-IGNITION

The engine was running on producer gas, and at first I thought the gas might be causing the trouble, as it will, if very rich. Another cause might have come from the piston getting hot, as it was noncooled. After a careful investigation I found it was due to the spark plug projecting too far into the cylinder and, being exposed too much to the flame caused by the explosion, it would, after a time, collect a layer of carbon on the plug end, and the carbon being kept at a glow by the continual explosion, fired some of the charges before the proper time. I had the body of the plug made shorter by turning off the end, as in Fig. 2, to allow the plug end and contact breaker to be slightly recessed, about ½ inch below the surface of the cylinder, so to be quite free from the flame. On restarting I found that I had overcome the trouble.

I experienced pre-ignition on another engine in quite another way, which was caused by a small flame in the hole used for indicating, Fig. 3. When not indicating the hole is stopped up by means of a brass plug, shown at *B*.

When the plug is screwed down a small space *C* is formed between the bottom of plug *B* and top of hole *D*. I overcame this by decreasing the area of the hole *D* by the means of a piece of brass tubing.

The best way would have been to drill a hole in bottom of plug *B*, and screw in a piece of brass rod of sufficient length to come flush with the inside cylinder wall. By so doing the hole is entirely filled, thus preventing any gas collecting in the hole, and doing away with the sharp edges, which are very bad, and if very fine will cause pre-ignition by getting red hot.

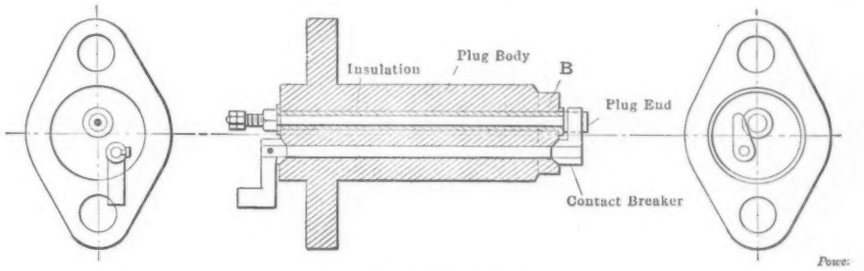


FIG. 2. SPARK PLUG

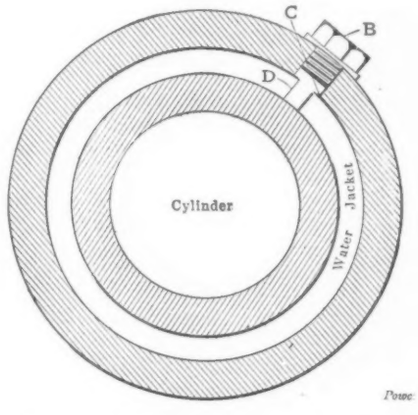


FIG. 3. CROSS-SECTION OF CYLINDER

FRANCIS H. FIELDING.

Manchester, England.

FROM·READERS·WITH SOMETHING·TO·SAY

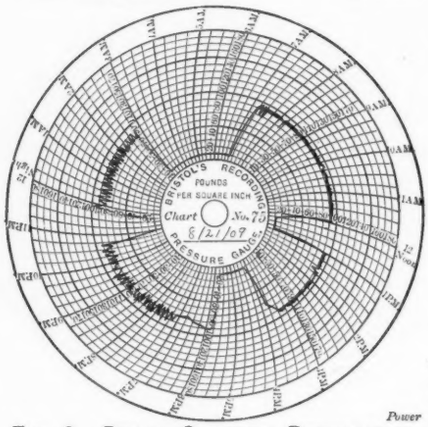
Practical Information from the Man on the Job. A Letter good enough to print is good enough to pay for

Air Pressure Regulator

The accompanying sketch is of a spring device which may help some engineer who is bothered with varying loads. I have applied it to the regulator or cutout of a 25 and 15 by 20 compound air compressor, 135 revolutions per minute and driven by a 200-kilowatt 3-phase, 60-cycle, synchronous motor running at 600 revolutions per minute.

The chart, Fig. 2, shows the variation in pressure without the spring, from 7 p.m. till 3 a.m., night shift in the mine, not much air being used. I put on the spring after starting at 7 a.m. and after regulating the length of the center bolt the pressure of the air was steady at a pressure of from 90 to 95 pounds, as will be seen on the chart, the spring keeping the inlet valve open just enough to hold the desired pressure. It has been in use now over two months and needs no attention.

This sluggish action of the regulator is, I think, due to the fact that there is 31 feet of ¼-inch pipe with four elbows, connected to 35 feet of 1½-inch pipe with five elbows, before connecting to the air receiver.



Power

FIG. 2. CHART SHOWING REGULATION

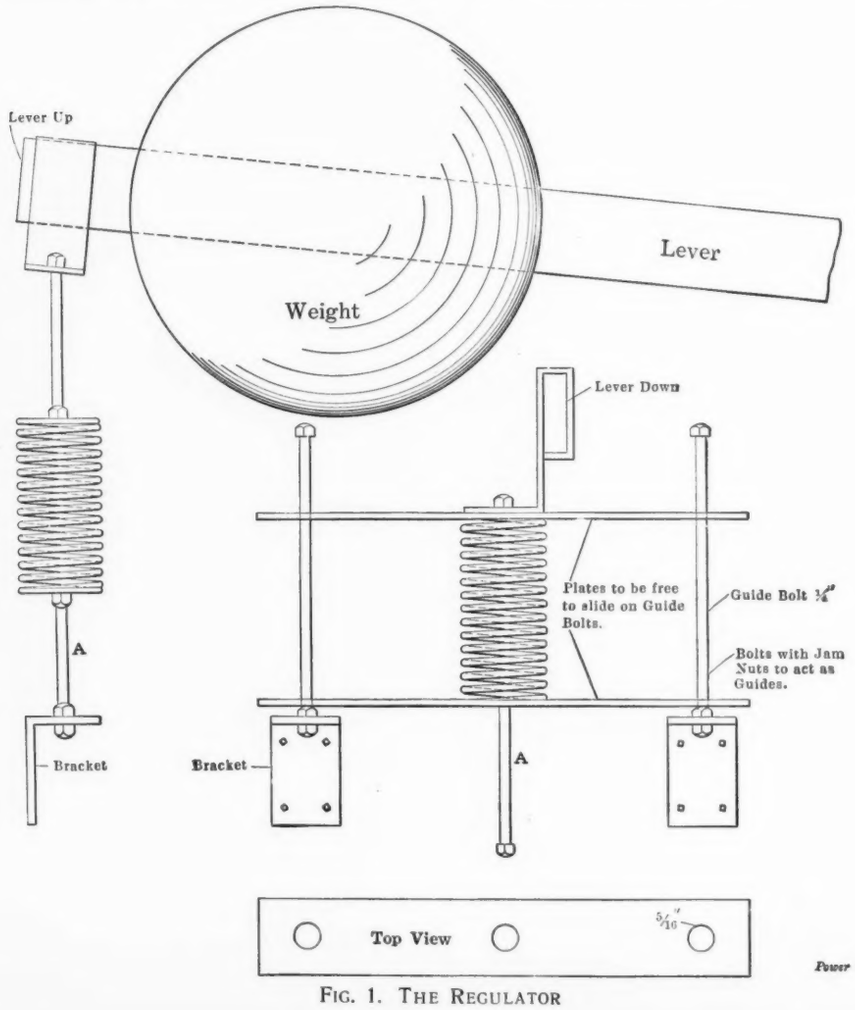
The center bolt *A* must be long enough to allow the lever to go up to the end of its travel and have a good tension on the spring tending to pull it down. This keeps the lever floating and makes it follow any variation in the air pressure instead of staying in a closed position until the pressure falls 10 pounds before starting to open the inlet valve. The top plate must compress the spring far enough to allow the lever to come down to the end of its travel, so the inlet valve will be wide open.

The regulator does not go from one extreme of its travel to the other, but follows the air pressure, keeping a nearly constant load on the motor.

Two springs may be used, one on each guide bolt, if it is more convenient to fasten the device on the fulcrum side of the weight. I used a very light spring made from No. 14 Brown & Sharpe gage steel wire.

WILLIAM BEATON.

Gold Roads, Ariz.



Power

FIG. 1. THE REGULATOR

A "Swelled" Cylinder Head

In a power house not far from Worcester, Mass., is a cross-compound condensing engine, 24 and 48 by 48 inches, running 110 revolutions per minute. For about two years past, it has been noticed, when removing the back head of the low-pressure cylinder, that it was very tight in the counterbore, and before it could be replaced the head had to be reduced by filing on the counterbore fit. A short time ago, when the head was removed, it was found to contain several fractures on both the cylinder and outer sides. One fracture, 24 inches long, was found inside the cylinder running lengthways with the bore, just above one end of the exhaust port, as shown at *F* in Fig. 3. Fractures were found through the bridges of the exhaust ports shown at F_1, Fig. 3. Fractures were also found through the bridges of the steam ports at *E*, Fig. 1. This cylinder is fitted with steam jacketed heads and a pressure of from 140 to 150 pounds jacket pressure is used, and it is supposed that the alternate expansion and contraction of the head together with

the jacket pressure caused it to retain a permanent set sufficient to cause a bursting pressure in the counterbore and in turn cause the fractures indicated. That excessive pressure existed between the head and counterbore was shown by the black, shiny appearance of the metal the depth of the head, such as can usually be noticed where great pressure is exerted between two machined cast-iron surfaces well lubricated. Six jack screws

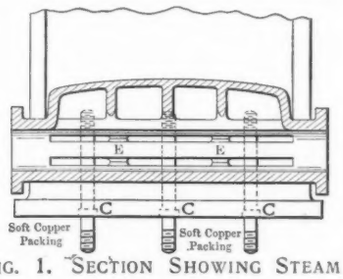


FIG. 1. SECTION SHOWING STEAM PORT

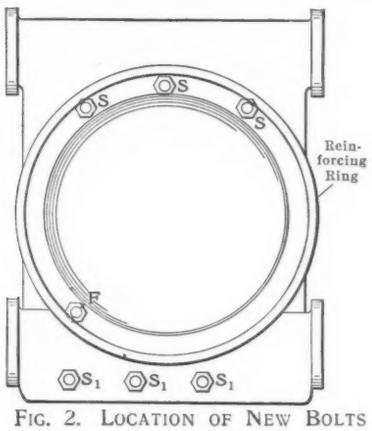


FIG. 2. LOCATION OF NEW BOLTS

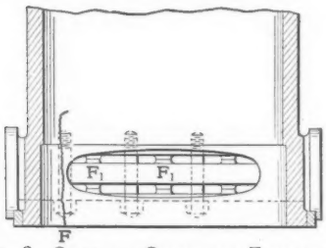


FIG. 3. SECTION SHOWING EXHAUST PORT

and sometimes as high as 30 pounds of steam pressure were required to remove the head.

To repair the damage, a boring bar with a specially constructed head was used to turn down the outside of the cylinder flange, reducing the diameter 1½ inches. The cast-iron casing which covers the cylinder and acts as a support to the head end by engaging the cylinder flange was bored ½ inch larger. A steel ring 1 inch thick, 2 inches wide and 4 feet 7½ inches in diameter was then made and shrunk over the flange, ⅛ inch being allowed for shrinkage. To heat this ring a large gasolene torch with two burners was used. To prevent the ring changing its shape while being heated and the better to handle it, a harness of 1x⅜-inch iron was made, and secured to the ring by clamps at six equally spaced points. The ring was suspended horizontally and accurately balanced on the hook of the station crane. This hook having a ball bearing was revolved very easily, while the gasolene burners applied heat at opposite sides. A gage was made of the cylinder flange and as soon as the ring was sufficiently expanded, it was hoisted up, with a rope falls, placed in position and allowed to cool.

Three long studs were put through the steam ports, two of them, 1⅜ inches in diameter, taking the place of two regular flange studs and one 1¼ inches in diameter was located between the regular studs at a point low enough to clear the bottom of the valve port, shown at *S*, Fig. 2. The holes for these studs were enlarged ⅛ inch at the flange end for a depth of ⅜ inch and when the studs were in place, soft copper wire was calked in around them, as shown at *C*, Fig. 1.

Three 1¼-inch studs S_1, Fig. 2, were placed through the exhaust port so as to come below the valve seat. These studs when being tightened up were heated with a plumber's torch, allowing them to expand; a soft-copper washer was placed under each nut as a gasket. A hole was drilled at the end of the fracture in the counterbore to prevent it spreading, and plugged with round copper, calked and riveted to prevent leakage.

The cylinder head was machined on the counterbore fit and flange, and the steam jacket removed.

W. H. CYMRY.

Providence, R. I.

Putnam Engine Governor Adjustments

In order to get the maximum amount of power from the Putnam engine, it is necessary that the governor be given its entire range of working. To do so, put the balls in their higher position, and turn the cam shaft, throw in or out the bell crank *I*, by loosening the set nuts and taper pin on the shaft. In some engines an eccentric pin at *I* will do the same thing until the cams do not touch the levers. Then lower the governor to its running position and move the crosshead out and see when cutoff takes place. Make a mark on the guides and do the same to other end, and if not alike advance the cam for earlier cutoff, or throw it back for later cutoff. This method is to be followed only after the setting of the valves

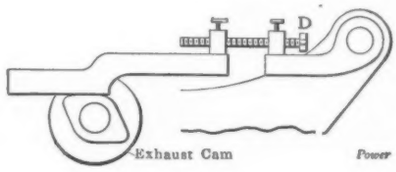


FIG. 2

by the regular rule or as a final adjustment, as advancing these cams affects the lead. The adjustment of the governor must be given especial attention, because without it, the range of cutoff will be very short and the range of cutoff may be decreased if the steam cam *A* does not strike the cam lever flat, as shown in Fig. 1, when the engine is on the dead center. In so doing all of the upward movement of the cam will be available in keeping the valve open longer with a given position of governor, and any advancement of cam *A* due to excessive lead is sure to decrease the range of cutoff.

The wear of the brass block *B* is considerable due to the sliding in and out of the cam lever in changing the cutoff, and allowing the lever to rise, due to the spring *I*. Unless a new block is put in or shims under the old block, the cam will be later in meeting the lever, and the port opening will be slower, showing a leaning admission and wire-drawn steam line. Frequently this block will fall out entirely if the spring *I* gets weak and the valve will hardly open at all.

The leaking of the poppet valves is due to the unequal wear of the top and bottom, there being less metal in the bottom than top; and the top poppet will keep the bottom one off the seat, hence the bottom one will leak. This is also true

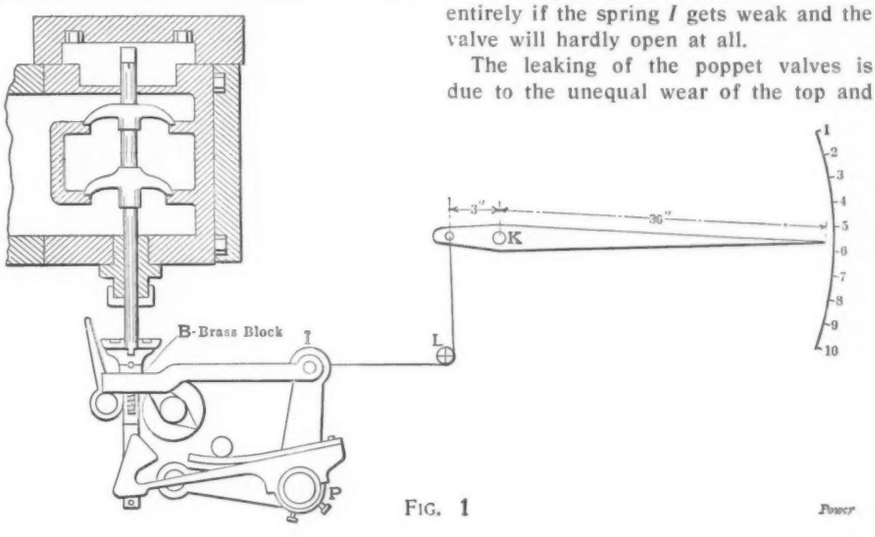


FIG. 1

Power

of the exhaust valves. To make tight take the valve out and put an equal amount of flour of emery on each seat and then take a screwdriver and turn the valve around until a tight seat is obtained. Any hunting of the governor can be stopped by tightening the friction bearing at *P*, care being taken not to make it act sluggish. There is another friction bearing on the cam shaft, which is used for taking up the back lash in the gears at both ends. Experience has taught that this type of engine does not get rid of a dose of water as readily as other valve gears such as the Corliss, Green or slide valves, because the only leverage it has is the difference between the under side of the top steam poppet and the top side of the bottom poppet valve, between which there is a very little difference. In view of these circumstances it is advisable to run with as little compression as possible, as in so doing the exhaust valve will be closed later in the stroke and less water will be entrapped. Again, if only a little compression is carried, it is almost impossible to back up the engine when beyond ¾ stroke, as, when lifting the steam valves with a bar to back up, the exhaust valve is open at the same time on the same end, hence, with a heavy friction load it is a very difficult thing. With early compression the exhaust valve is closed early in the stroke and the opportunity to get a head of steam to back with is better.

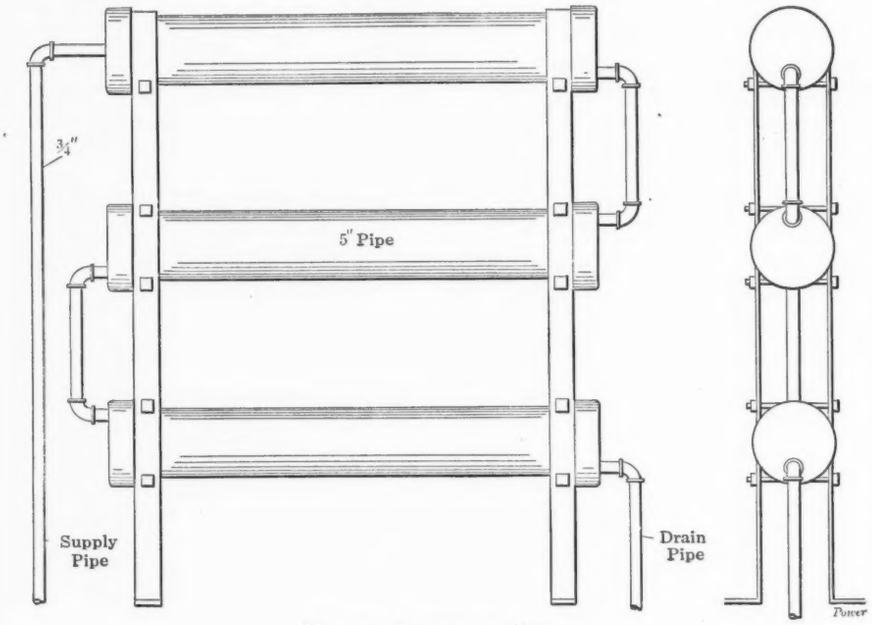
When running these engines condensing and with a compression that is too high for noncondensing, when the vacuum is lost there is a tendency to pound. In Fig. 2 the exhaust lever is in two pieces coupled together with a right- and left-threaded bolt *D*, with check bolts on top. By turning this bolt the bearing surface of the lever is lengthened or shortened and so governs the time of closing the exhaust port. Therefore, the compression may be made to suit the conditions by having marks on the bolt, and at the same time it does not affect the release. This change can be done while running. A simple cutoff telltale is easily made by attaching a fine wire on the steam bell crank *I* and then to a pulley *L* on the wall, then up to a lever with a fulcrum at *K*. It is very interesting in noting the change in cutoff while adding back pressure, as the effect is greatly magnified.

A. C. WALDRON.

Lynn, Mass.

A Heater from Scrap

The accompanying sketch shows a steam heater made mostly from scrap and installed in our station office. It consists of three pieces of 5-inch pipe 3 feet long with the ends capped and tapped for ¾-inch pipe which connects the whole together in the manner shown. The stand is of ⅜x2-inch strap iron bent to form shown, and held together by bolts running through immediately over and under the 3-inch pipe. Feet are formed at the ends and drilled for screwing to the floor. The number of sections could be increased to suit local conditions.



HEATER BUILT OF SCRAPS

By painting the stand and parts black and the small pipe, caps and fittings with aluminum paint, a very pleasing effect may be obtained. Valves, not shown in the sketch, in the supply and drain lines regulate the quantity of heat given off.

C. L. GREER.

Handley, Texas.

Lifting Bolt Hole Forms

The wooden form used in making bolt holes in concrete work is a stubborn article to deal with when it is to be removed from the hole. Usually the forms are made from rough lumber, the surface of which tightly grips the surrounding concrete. Nails, knot holes and checks in the boards add to the strength of the adhesion.

An extremely simple and highly efficient device for gripping the form consists of a piece of pipe slightly larger than the internal width of the form, and long enough to allow a grip on the form consistent with the effort required for its removal. A coarse thread should be cut on one end of the pipe and a hole drilled through the other end for attaching a clevis or receiving a bar. By means of a jack and lever arranged as shown in Fig. 1, the form may be removed. Where an overhead crane is installed, the simplest method is to fasten a clevis on the lifting pipe and let the crane do the work. Regarding the strength of the grip between the pipe and wood, I know of an instance where the heavy lift of a 30-ton crane was required to extract the form for a 3-inch bolt, set 6 feet in the concrete.

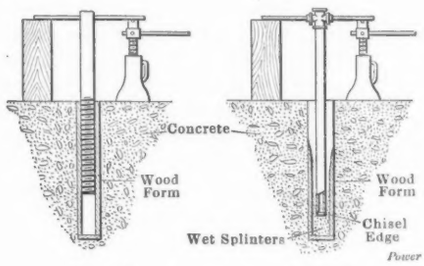


FIG. 1 FIG. 2

When dealing with refractory forms, by far the most irritating condition of affairs is to have the form become waterlogged. In this case it is impossible to make the lifting pipe hold; and if the hole is deep, practically nothing can be accomplished with a chisel bar, except to plug the bottom of the hole with a spongy mass of soggy splinters. The only way to remove these is by means of a pipe which has a chisel edge on its lower end obtained by filing the inside edge of the pipe with a rat-tail file. Drive in the pipe with a sledge and lift it out with a jack. The splinters rammed in the pipe may be driven out with the handle of a crowbar. After a few drives the core of the soggy mass will be so far removed that the sides may be drawn out with a hooked bar. In Fig. 2, showing this arrangement, the cross on the top of the pipe is convenient, but not necessary. The lever may be secured to the pipe with a chain or rope hitch.

J. J. O'BRIEN.

Buffalo, N. Y.

Arranging Condensation Returns

The extravagance which lack of forethought in the disposition of available apparatus entails in many steam plants, and the saving that may be effected by a rearrangement according to correct principles, are strongly exemplified in the case of the equipment illustrated in the accompanying illustrations, where Fig. 1 shows the plan adopted for returning the water of condensation from a number of high-pressure steam lines, and also the method employed for disposing of the condensation in the exhaust-steam heating coils of the hot-water supply tanks, as originally designed and installed in a public building. Fig. 2 shows how the same work was performed upon a more economical basis, without the installation of any additional apparatus, but by the exercise of good judgment in utilizing the equipment already at hand.

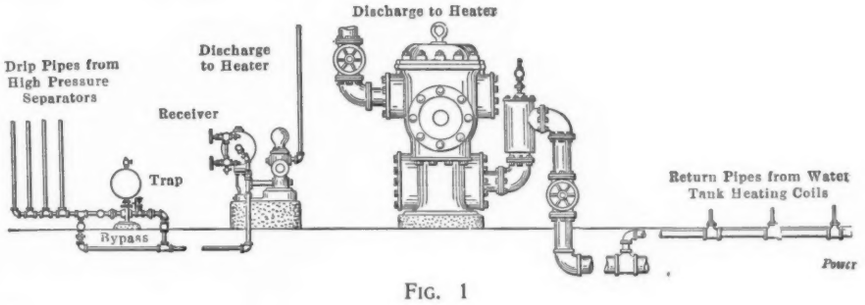


FIG. 1

In the matter of draining the steam lines, the scheme of arrangement, as originally outlined, contemplated the accumulation of the drips from four high-pressure steam separators in a single trap of the tilting variety; but as the distance intervening between this trap and the feed-water heater that was destined to receive its discharge was considered excessive, it was deemed advisable to provide a small tank as a receptacle for the water discharged by the trap, from which it could be delivered to the heater by a steam pump.

With regard to the exhaust-steam coils in the hot-water supply tanks, these were a part of a vacuum heating system installed in the building. During the summer months, the only portion of the heating system in service comprised these coils, but as no provision had been made for a vacuum pump of a size commensurate with the work of pumping out these coils, the duty naturally devolved upon one of the big pumps, which was ample for the entire heating system extending through a building ten stories high, and covering an area of one city block.

This manner of arranging the outfit struck a new engineer who came on the job as being, in the one case, a useless elaboration, and in the other a complete contravention of all established notions respecting steam-engine economy; so he decided upon a readjustment of the apparatus in the manner shown in Fig. 2.

The discharge from the trap was piped directly to the heater, no trouble or inconvenience whatever being experienced on account of the length of the line, while the little pump thus placed at liberty was used for pumping the returns from the supply-tank coils.

The most obvious result of the altered arrangement was a marked economy in fuel consumption during the summer months. This was an outcome only natural to expect, for it is, manifestly, quite reasonable to assume that the steam which had formerly been consumed in overcoming the inertia of the reciprocating parts of the larger pump and keeping them in motion, was really many times in excess of the total amount consumed in actually pumping the condensation from the heating coils to the feed-water heater in the case of the smaller pump.

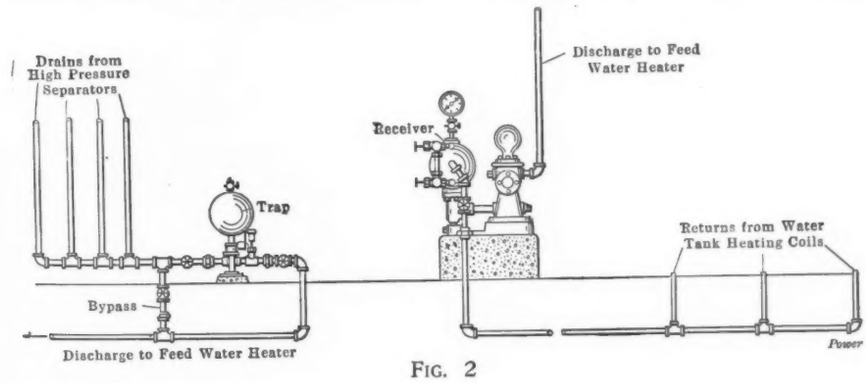


FIG. 2

A. J. DIXON.

Chicago, Ill.

Running a Large Engine at High Speed

Sometime ago an accident happened in my plant in which the only generating unit and one of the heavy-duty mill engines played an important part. The engine of the generating unit, a 20 by 86-inch Corliss was quite badly wrecked by what was supposed to be a broken follower plate, the piston being of the built-up type. The damage to the engine was a broken piston, broken and badly bent rod, cracked cylinder head, broken crosshead shoe, broken metallic packing, a piece broken out of the packing gland and both exhaust valve stems badly twisted.

An examination showed that the follower plate had an old crack running nearly one-half way through it. It was known that all was not exactly right with the piston, as several follower bolts had been broken at different times. All, with the exception of one, went through the exhaust valves, the one being punched through the cylinder head, leaving a hole about 1½ inches in diameter, but not cracking the head, which was tapped out and a plug screwed in.

As the plant was behind on orders and another unit was soon to be installed, the engine was not shut down for overhauling, until it did so of its own accord.

As the engine would be down several days it was necessary to devise some way of getting electric power for the machine shop and mill. This was done by moving two 50-horsepower motors so they could be belted to the jack shaft of one of the 32 by 48-inch mill engines having piston valves and a 16-foot flywheel, grooved for twenty 2-inch ropes. It ran at the usual speed of 90 revolutions per minute.

Two wood pulleys were made and turned up in a machine shop for four ropes each, and keyed to the motor shafts. They were supposed to be the proper size to give the right speed to the motors, which were to run as generators, when running the engine 100 revolutions per minute; but a mistake was made in turning which was not noticed until the engine was started, when it was necessary to increase the speed to 128 revolutions per minute. The speed was later reduced to 118 revolutions per minute, by turning the pulleys smaller. The generators were connected to the switchboard in parallel by laying about 250 feet of cables over the ground, and a shunt-field line from the switchboard, using one rheostat to control the fields of both generators; and in this manner current was furnished for about one-half of the mill until the other engine was ready to run. The flywheel when running at the highest speed had a rim speed of 6430 feet per minute, which is above the safe speed for

a cast-iron wheel of this size. This wheel had two rows of eight spokes each, and eight 1¼-inch rods from the rim to the hub. The engine was run in this manner about 10 days until the other engine was ready for service and gave no trouble, although a 100-horsepower load on a 900-horsepower engine was a little out of proportion.

Some trouble was expected with the generators as one had shunt and the other compound fields, but by cutting the series field out they worked nicely.

J. C. HAWKINS.

Elyria, O.

Combustion Qualities of Coal

The qualities of fuel burned under the boilers are several, and each is classified for the generation of steam according to its merits. For instance, Pocahontas is considered as the best, if burned with a mechanical device or a stronger draft. New River coal is also regarded as the best when it is burned with natural and weak draft; 50 per cent. of its production is consumed in the New England States. It has a high percentage of volatile matter and low percentage of ash, and it ignites quickly and with freedom. On account of its prompt ignition the temperature in a furnace will be higher than with any other fuel; the slicing and hoeing of the fire also will be less, because the coal does not need so much handling, therefore the percentage of ash falling in the ashpit will be less, compared with the Pocahontas.

Pocahontas coal is also considered by some to give better results than the New River, when the conditions permit. In the writer's experience in testing the efficiencies of boilers with different grades of fuel, it was found that the New River coal gave a more constant steam pressure, and made less work for the fireman than Pocahontas, but more economical results were obtained with the latter than with the former. This occurred when the boilers run at their rated horsepower, but when the percentage of rated horsepower increased, the New River was the best, as it generated the steam quickly; whereas the Pocahontas, on account of its slower ignition, necessitated harder labor and more manipulation of the fires, causing the fuel to fall down in the ashpit. With this coal a larger percentage of air space is required, as compared with New River; the smoke is also less dense than the latter, on account of its lower volatile matter. In the writer's evaporation tests it was found that Pocahontas gave 5 per cent. better results in economy than the New River coal with mechanical draft.

Hard-coal screenings, with mechanical draft, will give better results in economy than No. 3 buckwheat or birdseye. This fuel, on account of its components, does not burn as freely as the buckwheat or birdseye, and, therefore, needs more manipulation of the fires by slicing and hoeing. When this coal is mixed half and half with soft coal, it retards the ignition of the latter, therefore the mixture will burn slowly, causing a fusion of both; this is caused by the natural draft not being strong enough to allow the air to penetrate the fuel. The free-burning buckwheat does not need slicing because its clinkers are porous, and air will readily pass through. If the fireman is ignorant as to the method of burning it, then it will give just as much trouble as screenings do.

In the writer's experience in testing several plants for their efficiencies, having horizontal tubular boilers with natural draft, it was found that when clear Pocahontas coal was burned the average evaporation per pound of coal from and at 212 degrees Fahrenheit was 10.5 pounds in 24-hour tests, and when the same coal was mixed with half and half of hard-coal screenings, the evaporation was reduced to 8.5 pounds. The Pocahontas contained 14,500 B.t.u., the screenings 12,500 and the mixture 13,500; the latter producing an evaporation of about 9.5 pounds, and a boiler efficiency of about 60 per cent., whereas when all Pocahontas was burned the boiler efficiency was 70 per cent. This shows that when the above mixture is burned under a boiler using natural draft, the efficiency or evaporation is reduced proportionately according to the coal mixture burned.

Why is the efficiency of the boiler or furnace reduced when a mixture is burned with natural draft? It is on account of the screenings being hard to burn with the natural draft and, therefore, the soft coal must assist, and by doing this sacrifices some of its heat; thus the temperature of the furnace is reduced and the automatic damper kept open wider and longer, allowing air to pass through the furnace and out to the chimney, taking with it a large amount of heat.

If, however, this mixture is burned with a common forced draft, the air which will be driven through the fuel will be excessive; if the forced draft is installed with the same grate as the natural draft, it will not give any better result as to economy than with natural draft, because three times as much air will be driven through the fuel, therefore the surplus air will carry more heat units out of the chimney than natural draft.

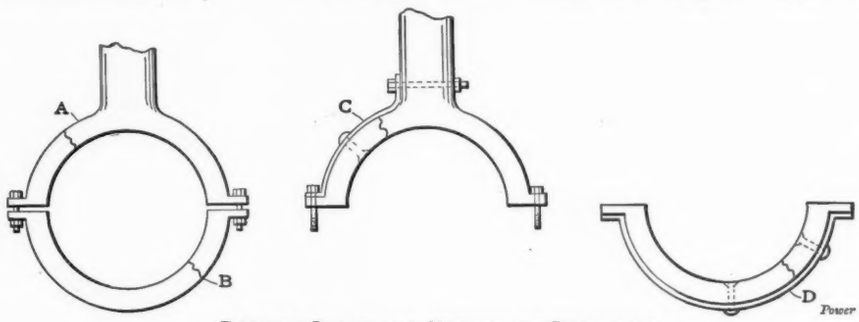
V. MACKAY.

Boston, Mass.

Repairing a Broken Eccentric Strap

While working for a contractor a couple of years ago I ran two rock crushers with a compound double-eccentric marine engine of 45 horsepower; it was a second-hand engine, and pretty well worn. One day one of the eccentric straps broke in two places, shown at *A* and *B*.

As I could just about keep things going by running 10 hours per day, we could not wait to have a new strap made; so I took it to the shop, drilled a hole through the shank, also through the eccentric strap, and forged a piece of



BROKEN STRAP AND METHOD OF REPAIRING

1x¼-inch steel so it would fit as shown at *C* and *D*. Holes were drilled and bolts put in place. The holes on the inside of the strap were countersunk, so as to form a smooth surface. We ran this repaired strap for over six months and had no trouble.

W. ELLERBROCK.

Honolulu.

Belt Runs Off Pulley

I have been having trouble with a 10-inch belt driving a 120-kilowatt alternator, 1100 volts and 80-ampere load. The belt slipped a good deal, but never ran off the pulley until one night during a storm, when the machine got a bolt of lightning that threw the belt over against the side of the machine, and stretched one side more than the other. Since then we have had considerable trouble to keep the belt on.

The belt will run all right as long as the load is heavy, but as soon as the load goes off the belt does likewise. It is 91 feet long, with 32 feet centers, driving pulley 14 feet in diameter and the driven 16 inches.

JOHN BERKLEY.

Homer, Ill.

SUBJECTS UNDER DISCUSSION

Comment, criticism and debate upon various articles, letters and editorials which have appeared in previous issues

Cylinder Lubrication

In working out engineering data it is most essential to reduce them to some convenient form and plot the results. It is seldom that the data offered will be just the one result which is desired, and by means of the plotted diagram any result may be obtained, and at the same time it forms a check on the accuracy of the observation. Anyone who has followed the letters on lubrication will realize of how little value are unclassified data. If perchance one set of conditions happened to fit the conditions in question there is no means of checking the accuracy of the given data. However, if several results could be compared more definite conclusions could be obtained. In order, then, to check the data, and also furnish a means to make those data applicable to any engine, the following curves have been plotted.

The coördinates which have been chosen are surface lubricated horizontally and pints of oil used per 10 hours vertically. The data can be easily reduced to this form and the results are shown by the curve. To read the curve for any engine it will be necessary to find the circumference of the bore and multiply that by the length of stroke in inches. In other words,

Surface lubricated = *diameter* × 3.1416 × *the stroke.*

Taking a 26 and 42 by 40-inch compound engine, the calculation would be

$$26 \times 3.1416 \times 40 = 3267.26 \text{ square inches,}$$

and

$$42 \times 3.1416 \times 40 = 5277.89 \text{ square inches,}$$

$$3267.26 + 5277.89 = 8545.152 \text{ square inches,}$$

or, say 8500 square inches.

This engine would then use about five pints of oil in a 10-hour run. The way in which this was obtained is shown by the light line on the curve, Fig. 1, for compound engines. Any engine may be checked as to the quantity of oil used in this way, and the curves give a ready means of showing how much variation can be expected.

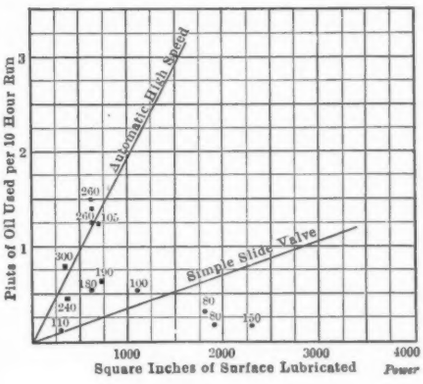


FIG. 3

As plotted the points along the curves show quite a uniformity, and when the great variety of conditions is considered from which they come, the results, especially in the case of the compound engines, show considerable regularity. On account of this regularity it is quite evident that there are some very necessary conditions which must be fulfilled in order to have good lubrication. If there were no such conditions the points would fall all over the paper. The running engineer has, in perhaps the majority of these cases, tested out the oils until he finds one that is suited to his purposes, and hence, as far as quality of oil is concerned, the results will show little variation.

The average curve, as plotted for the Corliss engines, comes remarkably close to the curve for the compound engines. However, a compound engine usually operates with a four-valve valve gear and, as far as lubrication goes, is practically a double Corliss and, hence, does not require any different lubrication.

On the curve sheet, Fig. 3, are shown the results for the slide-valve engines, the simple slide valve and the high-speed automatic engine. There seems to be a kind of shading off from one to the other, and there are hardly enough points to establish any definite curve. The simple slide-valve engine being simple and hard to get out of order receives less attention and is subject to many more abuses than the more expensive engines; as a result, much less care is taken in regard to lubrication, and hence the data do not show much consistency. The high-speed engine requires more oil than any of the other engines, and while this may be due to the higher speed, it seems probable that there is something about the valve gear of a high-speed engine that seems

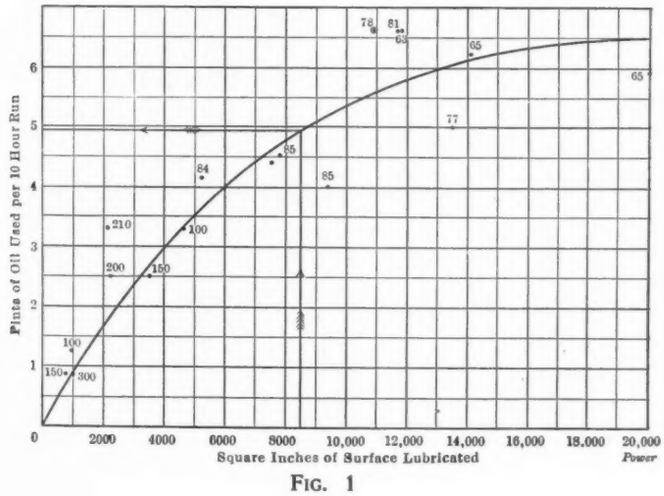


FIG. 1

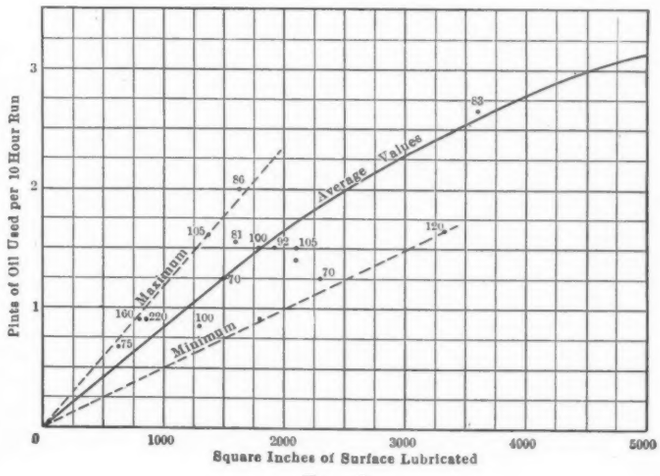


FIG. 2

to cause trouble unless the engine be properly lubricated.

A comparison of all the curves seems to show some important results, although they offer but little help in the solution of lubrication difficulties. However, it is just these points that make the engineering profession interesting. Much of the value of experience of an engineer would be lost if it were not necessary for the engineer once in a while to settle down and work himself out of a difficulty.

On nearly all the points plotted the revolutions per minute are given, and in the majority of cases the speed does not seem to play a very important part, although, as a rule, the higher speeds seek the part above the curve, showing the higher speed needs a little more lubrication.

Boiler pressure probably enters the problem to an even less extent than the speed, although there is no proof one way or another. The kind of oil used has probably a large effect, but when a plant is operating day after day the engineer tries out oils until he finds a suitable one. There are undoubtedly several oils which satisfy the conditions and are probably very much alike and, therefore, it is quite probable that the data already published have been obtained with the use of similar oils.

J. H. SPOOR.

Madison, Wis.

Tension in Cylinder Head Bolts

Fig. 1 shows a framework in which the lower horizontal member is held against the ends of the vertical strips by the tension of the scale spring which is drawn out until the index registers 30 pounds.

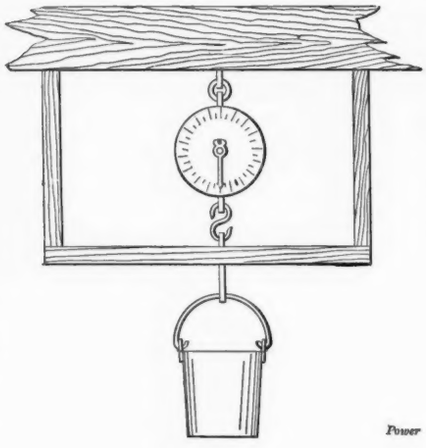


FIG. 1

To the under side of the lower cross member is attached a bucket into which water may be poured, the weight of the cross piece and the bucket being 3½ pounds.

Between the upper cross piece and the top end of the vertical members are strips of writing paper, which may be pulled upon from time to time as water is poured into the bucket. When thirty-six and one-half pounds of water are put into the bucket, the strips of paper, which up to

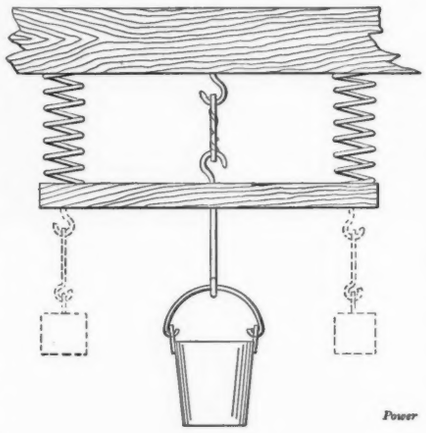


FIG. 2

that point have been fast, become free and can be moved, showing that until the initial tension to which the spring of the scale is stretched, is exceeded by the weight of the water in the bucket, no additional stress is put on it.

This accurately represents the condition of flange and cylinder-head bolts where there is no elastic gasket between the flanges.

For flanges having elastic gaskets interposed the condition may be represented by Fig. 2, where the rigid member is a length of wire extending from one cross piece to the other and in which tension is secured by the compression of the springs. The tendency to produce tension in the wire is shown in the dotted weights.

As the tension in the wire is equal to the expansive effort of the springs, it is plain that any weight added to the lower cross piece, by putting water in the bucket, would increase the tension in the wire, which would be equal to that of the springs plus the weight of the water in the bucket, as may be shown by substituting a spring scale for the wire.

From this it may be concluded that with iron to iron faces or with inelastic gaskets steam pressure in the cylinder does not add to the tension on the bolts, while with an elastic gasket between the flanges the tension on the bolts is increased by the steam pressure in the cylinder.

GEORGE GORDON BISBEE.

Springfield, Vt.

In the October 26 issue of POWER AND THE ENGINEER, under the title of "Some Erroneous Ideas," there is a discussion of the question of the tension in cylinder-head bolts. And owing to the difference of opinion, many readers may have become interested in the matter; therefore, I wish to offer something which may be of general interest.

Referring to the figure, if the nut is screwed down, a tension is produced in the bolt, as shown by the arrows, the nut and bolt head tending to compress the metal in the flanges. It is a law of mechanics that every action must have an equal reaction. The reaction of the metal in the flanges due to resistance to compression is represented by the arrows. The metal in the flanges presses outward against the nut and bolt head with the same force that the nut and bolt head press inward against the flanges. When a body is subjected to a stress, no matter how small, a strain or deformation is produced in it, and this strain or deformation is proportional to the stress which produces it. If a pull of 1000 pounds stretches a body 1/16 inch, a pull of 2000 pounds will stretch it ⅛ inch, providing the force of 2000 pounds is within the elastic limit of the material.

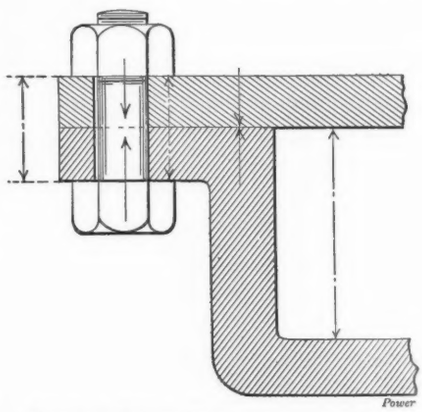
The tension causes the bolt to stretch, and at the same time produces a compression of the metal in the flanges.

When a working force comes into action tending to stretch the bolts further so that flanges will just be ready to separate, and as the bolt is already stretched due to the tension produced in it by screwing up the nut, and as this tension has also caused compression of the flanges, it follows that before separation of the flanges can begin, the bolt, already stretched, must be stretched still further.

In order that the flanges may be just ready to separate, the elongation of the bolt must be equal to the compression of the metal which is relieved.

In screwing up the nut a tension is produced in the bolt as shown by the arrows and their direction.

The reaction of the metal in flanges due to resistance to compression is also shown by other arrows and their direction. When the nut is first screwed up



ARROWS SHOW DIRECTION OF STRESS

and the pressure inside the cylinder is zero, the reaction equals the tension.

When the working force comes into action, tending to separate the flanges, then the reaction of the metal in the flanges against the nut and bolt head is

partly relieved, and when it reaches a point that the joint is about to open the reaction is totally relieved.

The tightness of the joint depends upon the elasticity of the metal in the flanges.

Anything which increases the total compression increases also the tightness of the joint. This may be accomplished by increasing the thickness of the metal compressed, by increasing the total compressive stress, or by decreasing the area of the metal compressed by the bolt tension.

As these methods of increasing the tightness of the joint are not commercially practical, the usual method is to introduce an elastic gasket between the flanges.

It follows that the tension in the bolt when the joint is about to open must be greater than the initial tension due to screwing up the nuts.

Therefore, if the working force or pressure be limited to a value equal to or less than the bolt tension, there will be no opening of the joint.

It also follows that when the pressure equals the tension in the bolt the total tension is not the pressure plus the initial tension, but that due to the working pressure only, as the compression of the metal in flanges is now relieved.

As a general rule, in a metal to metal joint the area of the metal compressed by the bolt action is large compared with the cross-sectional area of the bolt, and the total compression or shortening of the metal in the flanges is very small compared with the total stretch of the bolt, so that the total tension in the bolt when the joint is about to open is not so much greater than the initial tension due to screwing up. To insure a tight joint the initial tension should be twice the pressure or greater.

After the bolts are stretched an amount due to the screwing-up tension, any further stretch of the bolt due to the internal pressure may not affect the initial tension to any great extent, as the pressure within the cylinder may not be great enough to cause an additional stretch in each bolt sufficient to relieve the compression in the metal flanges, or in the gasket where one is used.

If the elongation of the bolts due to the internal pressure is small compared with the compression of the gasket, the ultimate load on each bolt may approach the initial tension due to screwing up plus the tension caused by the internal pressure in the cylinder. This applies to a joint made up with a gasket.

Where rigid flanges are bolted together metal to metal, it is very probable that any further extension of the bolts due to internal pressure would relieve the compression of the metal in the flanges and the joint would be about to open and any increase in pressure would cause the joint to leak. It would then be "an erroneous idea" to suppose that the load on each bolt is that due to the initial tension plus that due to the pressure in the cylinder.

When the steam pressure comes into action tending to separate the flanges the compression of the metal in the flanges due to the initial tension is partly relieved, and when the pressure reaches a point where the joint is ready to open the compression of the metal is totally relieved and the reaction becomes zero.

The ultimate load on each bolt at this instant will be: The area of the cylinder cover exposed to the steam pressure times the pressure in pounds per square inch, divided by the number of bolts; or the load on each bolt will be that due to the pressure alone.

This is more liable to be the case in a metal to metal joint than where a gasket is used, as the compression of the metal in the flanges is small compared with the elongation of the bolts.

Where an elastic gasket is used between the faces of the flanges, the compression of the gasket is, as a rule, large compared with the elongation of the bolt, and where the connecting flanges are deflected, or sprung by the bolts, the case is similar to a joint made up with a gasket, exerting an outward thrust against the nut and bolt head.

The outward thrust of the metal in the flanges due to resistance to compression, produces tension in the bolt when screwing up, and when this is partly relieved the initial tension decreases, and at any instant the tension on the bolts must exceed the initial tension, but not necessarily reach a point equal to the initial tension plus that due to the pressure against the head. A careful study will convince one that in any practical case it would be difficult to decide what the ultimate tension in each bolt would be at any instant.

WILLIAM F. FISCHER.

New York City.

That the tension in the bolts holding a cylinder head in place is not increased by the steam pressure in the cylinder may be shown by the use of the apparatus illustrated in Fig. 1, where the lower cross piece is pulled against the end of the vertical members with a pressure of say 40 pounds by the tension of the spring of the scale. If water be poured into the bucket until the vertical members are free it will be found that the combined weight of the lower cross piece, the two vertical pieces and the bucket with its contents just equals the weight necessary to fetch the spring of the scale to the tension of 40 pounds; which shows that where no gasket is used or a gasket having no elasticity the tension on the cylinder-head bolts is not increased by the steam pressure in the cylinder.

Where an elastic gasket is interposed between the flanges the case is different, as may be illustrated by Fig. 2, where the bolts are represented by the wire and the gasket by the springs which impose a tension in the wire. It is plain from this figure that if additional pressure is put on the cylinder head, which pressure may be represented by the weights shown in

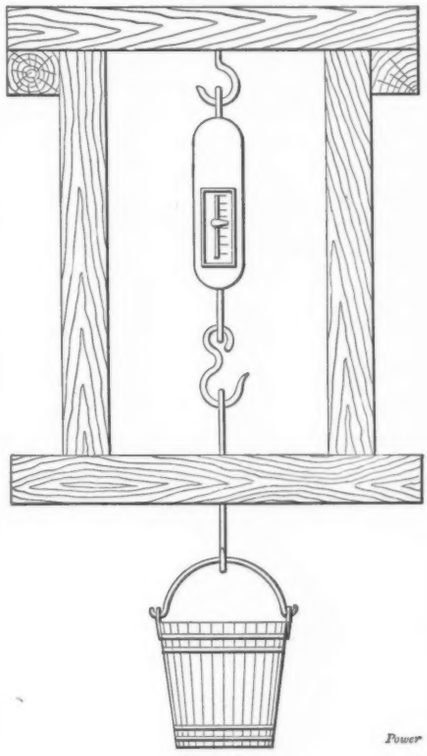


FIG. 1

dotted lines, this pressure will add to the tension already existing in the bolts. If the cylinder head and flange are inelastic and are held together without the interposition of an elastic gasket steam pressure in the cylinder will not add to

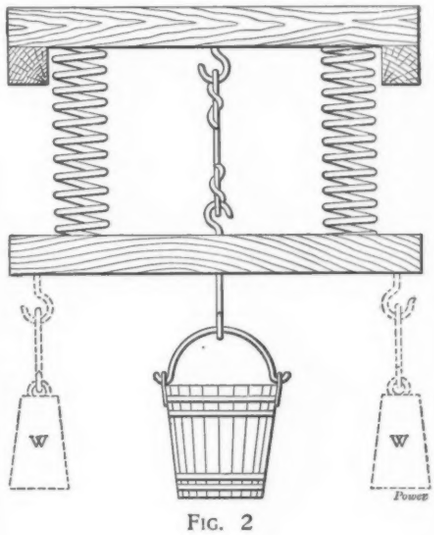


FIG. 2

the tension already existing in the bolts. If, however, an elasic gasket is interposed between the head and the flange, steam pressure in the cylinder will increase the tension of the bolts.

CHARLES HOYT.

Charlestown, N. H.

Economy of Superheated Steam in Steam Engines

Some time ago Mr. Johnson related two interesting experiences of the use of superheated steam in steam engines. It is quite evident that, in both cases, before the alterations were made, the engines were receiving wet steam, either due to priming in the boilers, or to condensation in the pipes, or, at best, steam that was merely dry. It has long been recognized that wet steam is a most uneconomical mixture with which to drive an engine.

Considering the matter from the very beginning, we know that the cylinder of an engine becomes alternately, at the beginning and end of each cycle respectively, hot and cold, using the terms relatively. The cylinder walls at the beginning of a stroke are much colder than the entering steam, therefore a certain amount of steam gives up its latent heat, in raising the temperature of these walls to that of the steam, and becomes water. It is obvious that this steam has done no useful work in driving the engine, and that its energy is, as far as the engineer is concerned, wasted. Now some of this water remains in the cylinder until the exhaust takes place, when, owing to the reduction of pressure, it immediately starts to boil, and escapes as steam. Part only, however, remains in the cylinder until the end of the stroke; the remainder leaks past the valve and is evaporated on the other side. This leaking is always taking place in ordinary slide-valve engines used without superheating, and constitutes a very serious loss indeed. The method of preventing it is quite simple, and consists in superheating the steam before it reaches the cylinder of the engine.

The superheating need only be of a comparatively slight degree to effect considerable saving. By this means the cylinder walls may be heated up by the superheat without any condensation taking place, and consequently all the steam does work upon the piston. Of course, condensation takes place during the stroke unless the superheat is rather high, but in no case is it as great as if the steam were not superheated. The saving by even a superheat of from 50 degrees to 100 degrees Fahrenheit reaches not uncommonly 10 to 20 per cent., and by superheating up to 300 degrees Fahrenheit a saving of 50 per cent., or more, has been made on the water consumption. The cost of the extra heat required is very little compared with the saving, as the specific heat of steam is small, being only about 0.48.

In the cases Mr. Johnson mentions the steam was superheated by throttling, and this superheating, although only probably a few degrees above the saturation temperature, was sufficient to prevent a certain amount of cylinder condensation, and so cause a saving to be made, in that less steam was required, and, therefore, less coal. The thing is very simple and yet it is one of those points which engineers sometimes neglect, and for this reason, as much as for the interest I have in reading Mr. Sawyer's adventures, I appreciate Mr. Johnson's article. Professor Ripper, of the Sheffield University, made some interesting experiments, about the year 1897, with a small Schmidt, noncondensing, superheated-steam motor, and he found that with a superheat of 300 degrees the consumption of steam per horse-power-hour was reduced from 38 to 17 pounds, a reduction of nearly 55.5 per cent.

CHARLES T. PIPER.

Sheffield, Eng.

Some Erroneous Ideas

In the issue of September 14 C. W. Hartley says:

"If a valve could be conceived underneath which there were no ports no degree of steam pressure would press it to its seat. If it were desired to move such a valve it would be found that it could be moved with equal ease whether in a vacuum or in a steam pressure of any intensity."

Of this statement J. B. Clapper in the issue of October 19 gives the following opinion:

"Strange how we've been plodding along all these years with the mistaken idea that steam under pressure will exert this pressure on all surfaces exposed to it. Engine builders have used balance plates, piston valves and whatnot, thinking that they were overcoming something, and now we are told there is nothing to overcome."

Strange how some men will twist things which they do not understand into all sorts of shapes and pretend to get meanings from plain statements of fact that could not be conceived by anyone who reads correctly. I did not say there was nothing to overcome. I said the balanced pressure on the back of a valve depends not on the area of the valve but on the area of the ports covered by the valve. To the best of my knowledge and belief this fact which needs no proof was discovered experimentally by Mark Lester, a Mississippi river steamboat engineer about sixty years ago.

He conceived the idea of balancing the slide valves of his engines by means of a diaphragm in the steam-chest cover to which the valve was attached by a short chain which swung like a pendulum as the valve traveled from one extreme position to the other.

He expected that the upward pressure on the diaphragm would balance the downward pressure on the valve, and reduce the power needed to move it. He made the first diaphragm of an area equal to the area of the valve with the result that the valve was suspended in the steam chest and would not touch the seat at all. He then reduced the area of the diaphragm to that of the area of the ports covered by the valve. As soon as the valve moved enough to uncover one of the ports the valve was again lifted from the seat, and it was only when the area of the diaphragm was reduced to the area of the ports covered all the time by the valve, i.e., the exhaust port and one steam port, would the valve remain on the seat throughout the stroke, proving clearly that it was the area of the open ports under the valve that determined the pressure holding the valve to its seat and to what extent it was pressed to its seat. Mr. Clapper further says:

"Let us consider a valve, as this writer suggests, that is doing nothing in the port-closing line, but is ground to a perfect bearing on its seat. We will say that the area of contact of this valve with its seat is 100 square inches, and there is steam pressure of 100 pounds per square inch; making a total pressure of 10,000 pounds on the valve. If instead of exposing it to steam pressure we were to put a weight of 10,000 pounds on it, does the writer of the article referred to contend that it could be moved on its seat as easily as before it was weighted? I have the erroneous idea that it could not."

No! A slide loaded with a 10,000-pound weight would not move as easily as one with no weight at all and the valve that does "nothing in the port-closing line" has no more pressure on its back than elsewhere. In fact it is balanced. If before bringing up false arguments which do not bear on the case at all he had tried the experiment I suggested with the Corliss engine valve he would never have written what he did. Finally he says:

"If there is any difference between the two cases cited, in so far as the pressing of the valve to its seat is concerned, I should like to have it pointed out. Again, anyone who has had experience scraping or grinding two surfaces to a bearing knows that it requires considerable force to separate them when the contact is perfect. This is due to atmospheric pressure on the surfaces opposite those in contact.

"Or, is this another erroneous idea?"

There is all of the difference in the world between the two cases. In one case the weight acts in one direction only; in the other the valve is entirely surrounded by the medium which presses equally in all directions. And if he will investigate scraped surfaces he will find that atmospheric pressure has nothing at all to do with the case. It is simply adhesion the force of which may be in-

creased or diminished by the interposition of different lubricants between the surfaces. If the ground or scraped surfaces which he mentions are wet with water it will be found that they will move upon one another with considerable friction, while if covered with oil they will slide freely but can be separated only by the use of considerable force.

The statements I have made are not guess work, but the statements of personal experience and observation, and when Mr. Clapper writes only of those things which he knows his communications will read much differently from what they do now.

C. W. HARTLEY.

Saugus, Mass.

Kerosene for Cleaning Refrigerating Coils

In the discussion of kerosene as a scale remover I want to mention a use to which kerosene oil can be used with the absorption refrigerating machine.

In using it it is convenient to have a three-way cock the size of the coils and a short piece of steam hose with a union at one end.

If there is but one machine, cut out one coil and take out the plugs at top and bottom. If the opening is horizontal on top, screw in a short nipple and elbow. Empty a small quantity of kerosene into the top of the coil until it commences to drip at the bottom; by introducing the oil slowly it has a chance to soak its way all around the inner circumference of the coil.

The three-way cock is connected to the steam and water supply with a common outlet connected to the steam hose. The union on the steam hose permits of connecting to the different coils without disturbing the other connections. After connecting to a coil, admit steam and water alternately, letting the steam run until the coil is fairly dry at the bottom, and water until it shows fairly cool. Blowing into a pail at the bottom will show how much scale is coming down.

If it is necessary to clean a condenser coil with a machine in operation, be sure to have the anhydrous as low as possible in the condenser. The steam passing through the coil in the anhydrous will evaporate it and cause the head pressure to rise possibly to a dangerous point.

I am using a well water that has large quantities of magnesia, I first tried a solution of soda ash circulated through one coil at a time for an hour, then blowing out with steam and water. I got the scale out, but got a hole in one of the coils.

Then the manufacturers recommended forcing small quantities of fine gravel through the coils with water or compressed air. As I had no air of more than 30 pounds pressure I used water. It was slow work, and slower still when I spent two nights and a day getting a "plug" of that gravel started.

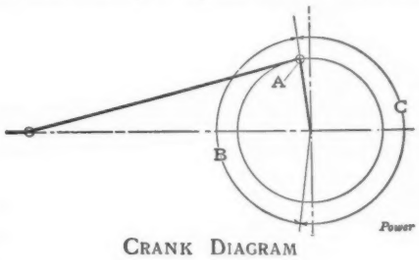
I am now using kerosene, and have very good results. I let the kerosene soak in for two hours and with fifteen minutes of the steam and water I have a clean coil.

W. C. REED.

Pittsfield, Mass.

Piston Speed Puzzle

Webster Tallmadge brings up the question of piston speed. Referring to the accompanying sketch, *A* shows the position of the crank pin when the piston is at mid travel; *B* equals the distance the



CRANK DIAGRAM

crank pin travels while the piston travels the back half of the stroke and returns to mid position; *C* equals the distance the crank pin travels, while the piston travels the front half of the stroke, and back to mid position.

It will be seen that the distance *B* is less than *C*, therefore it will take less time for the crank pin to travel *B* than it does *C*, which shows that the piston will travel faster at the back half of the stroke then it will at the front half.

P. C. ZIMMERMANN.

Hamilton, Ohio.

Does a Suction Head Help a Condenser?

The level at which a jet condenser is placed should be such that the vacuum which it maintains is more than sufficient to cause the condensing water to flow to it. Therefore, there can be no gain in taking water from an elevated point.

The surplus energy of which Mr. Thompson speaks, page 879, November 23 issue, does the same as the water going over the top of a dam or waterfall; it gets by without doing any useful work.

L. C. TUCKER.

Newburyport, Mass.

The condenser will require a certain amount of injection water, and whether it comes from a head or a lift will have no effect on the quantity used. If water is to be used (say alternately) from a head and a lift, the injection valve will have to be closed a little when drawing from the head, and opened a little when drawing from the lift.

The surplus energy of the head is used up in forcing the water through the opening of a partly closed value. The vacuum in the condenser is constant, the amount of water needed remains constant, and the lift or head varies. Therefore, some obstruction must be placed in the pipe that will use up the energy of the higher water levels.

R. MANLEY ORR.

Brantford, Ontario.

What Caused the Flywheel Explosion at Scranton

While reading Charles J. Mason's article, in the November 23 number of POWER AND THE ENGINEER, on the above subject it brought to my mind a little experience with a 10x12-inch high-speed engine, running about 300 revolutions per minute and driving a 50-kilowatt generator, 400 amperes. The engine was in all about the same size as the one spoken of at Scranton, and equipped with a Rites flywheel governor.

In changing over to put on a turbine-exciter unit, the load being removed, the engine started to run away, and before I got the throttle valve closed it was impossible to see the valve rod, driving rod, or any of the moving parts as the engine was running at a terrific speed. After shutting down, the first thing I looked at was the grease cup on the governor pin, which I found to be empty, and decided this to be the cause of the engine speeding up.

After filling the grease cup and forcing about a cupful of grease onto the governor pin, the cup was filled the second time and the engine again started; the load was thrown on, then taken off, this operation being repeated three or four times. The engine showed no signs of increasing its speed, governing itself perfectly.

This should be a lesson to all operating engineers having a high-speed engine in their charge, for the governor pin plays an important part in the regulation of the engine and should be kept well lubricated, as a dry governor pin will cause very poor regulation at times.

W. G. BARNARD.

Hartford, Conn.

A gasolene engine having the degree of compression ordinarily used for gasolene mixtures (about 70 pounds gage) will in general require 50 per cent. more denatured alcohol than gasolene per brake horsepower-hour. This is due to the fact that compression for alcohol mixtures should be carried to 180 pounds gage to obtain maximum results.

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DEVOTED TO THE GENERATION AND TRANSMISSION OF POWER

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Transmission of Heat

So far as known to science, heat energy is transferred from one body to another in two ways: by radiation, as a body is warmed by the sun or the blaze from an open fire; and by contact, as we are warmed or chilled by a draft of air blowing upon us. Within a single body itself, heat is also transferred in two ways: by conduction, from particle to particle in solid bodies; and by convection, in fluids whose particles are mobile and free to move amongst each other. In the typical case of a boiler setting with a fire ablaze in the furnace or firebox, the incandescent fuel bed and the volume of flame full of incandescent carbon are heating the shell by radiation: the extinct hot gases resulting from a complete combustion can only heat by contact with the metal. The outer layers of the shell heat the inner layers by conduction: the water lying close to the metal and warmed by contact with it becomes displaced by the cooler and heavier water seeking to find the lowest place, and by the currents due to this circulation a process of convection of heat is carried on, causing the mass of water to become heated uniformly throughout (or nearly so). Any factors tending to retard or prevent the rapid conduct of the radiation, contact, conduction and convection of the heat activity retard the transmission of the heat—or, in this case, the making of steam. The transfer or transmission of heat energy is always and normally downward on the scale of intensity of such energy; that is, the hotter or more active body tends to heat up the colder one. So far as known, the cooler body is never cooler after being exposed to conditions favorable to a transfer, nor is the hotter body ever hotter after contact or opportunity to give and receive heat by a radiation. This is true even in cases where the phenomena are those of refrigeration. It is the cooler mass of the refrigerant brine in the circulating pipes which is warmed by the masses to be cooled or frozen, and the latter are transferring their heat to the cooling medium. Hence a useful and convenient analogy has been much used, whereby the temperature or the heat energy are compared to a hydraulic head as of water in a vertical pipe. When that head is high, or rather when the difference of heat conditions of the two masses is great, then the transfer process is rapid and intense. As the heat condition equalizes between the two, the transfer process slows down and finally ceases. This simile is also suggestive of the existence of a thermal resistance of matter, a quality which causes the ease of transfer to vary either with some physical property of the body itself, or with some conditions of the location of the hotter and cooler body relatively to each other. So that if the problem be assumed to be that of the steam boiler in its setting, and the unit to be studied is assumed to be one cubic inch of flame volume in the firebox or combustion chamber, it will be apparent that in the transfer process such quantity of heat to reach the interior of the water mass will have to:

1. Radiate through all inches of distance between itself and the plate. Radiation is effective directly as the distance between the inch of flame and the shell; hence, if the flame can be kept hot enough not to go out by lowered temperature, the smaller the flame passages the better, when such flame passages are heating surface. So far as known, the radiating effect of any inch of flame is not interfered with by the activity of all other inches; hence, the bigger the flame the more effective for steam making. This explains why soft coal makes better steam coal than short-flame anthracite.

2. This radiating influence lasts until the cubic inch of gas is burned to invisible gas, or is so cooled in the transfer as to be no longer able to burn; or until the boiler shell is at such a high temperature that no further transfer from the flame can occur. The same cessation of transfer may occur under a fourth alternative, which is in the flow of the flaming gas from grate to chimney, the cubic inch under study had to leave the vicinity of receptive heating surface before it had had a chance to transfer all its available heat. Such heat is wasted up the chimney stack; the heating surface was not large enough, and a flue heater or economizer is suggested; or a thinner plate would have been better, allowing more rapid transfer during the time of the passage of that cubic inch of flame. In the first two alternatives, the cubic inch is now hot gas only and not flame, and heats

by contact. So far as known, hot gases carrying no flame do not radiate heat; or, if they do, it is at a negligible degree and rate. Now in contact transfers, experience shows that two resistances are offered at the surface of the receiving body. The first is a film adhering to the surface by a sort of capillary action, and acting as a nonconducting coating. This can only be swept away by strong draft in the gas, but is probably never completely eliminated. In the cooling surfaces of condensers (which are heating surfaces), this gas film or film of adhering water is most effective in retarding transfer, and heating and cooling tubes are more effective when placed across the flow of gas than when they are in the line of it, because in the former case the film is torn off more completely.

3. The other retarding influence is a nonmetallic layer of dirt or soot or tar or any mineral matter present mechanically to slow up the process of transfer in the time allowed. This applies to shell and to tubes.

4. Then comes the process of conduction of heat from the outer hotter layers of the boiler shell to those nearer the water. This is quite rapid and is uniform. And it is here that the thinner plate is of advantage because the thinner the plate, the more nearly are the outer layers at the lower temperature of the water as compared with that of the gas.

5. Then follows a contact transfer from metal to water within it, retarded as before by a lack of circulation speed, to keep the water film from adhering too long to the plate, and by the (6) presence of any mineral scale precipitated from the feed water.

7. Finally there is the convection process within the mass of water itself whereby any cubic inch of water therein receives to itself the heat transferred to the plate and reaching the films in (5) and (6) above.

While the foregoing analysis is not new to students of the problem, and will be found in certain textbooks in use in America, it receives especial interest at this time by reason of a most laborious and pains-taking research just completed in England and reported for record to the Institution of Mechanical Engineers by Prof. William Ernest Dalby, of the Central Technical School, of London. A very wide investigation has been carried into the bibliography of the subject, and abstracts of papers and experiments have been critically made and deposited in the library of the institution in London. These abstracts have been indexed, and such index is a paper to be incorporated into the transactions of the year 1909-10. The scope of topics covers combustion and flame, radiation, conductivity, circulation, heating surface, boiler-efficiency temperatures and superheaters. The sources are mainly European and continental, although work of the American locomotive-testing plants finds place under the class of efficiencies and special locomotive problems. No mention is made of some very interesting American work on treating the surface of metals with acid to modify the film action and prevent transfer of heat to metal within the engine cylinder. The late Charles E. Emery did some most instructive work with glass-lined cylinders at the old Novelty Iron Works of New York City, to obtain constants for such transfer. There is also no reference to the large body of literature in the United States on the transfer of heat through nonconducting coverings around steam pipes. A study of Professor Dalby's paper will be instructive to all interested, and is particularly to be commended as a model to those who may intend to conduct experiments in this (or in any other) field. The first step in well ordered experimentation is a modest and open-minded study of what others have already done.

The Government's Factor of Safety

At the present time, States and municipalities all over the land, are wrestling with bills and ordinances relating to the inspection of steam boilers. In such bills, the most important thing is the stipulation of pressure. This matter is well understood by all engineers worthy of the name, but time and time again, the efforts of these engineers to get a good bill in the matter of pressure are frustrated by some politician who will arise and address the meeting about as follows: "Gentlemen, I'm of the opinion, that what's good enough for Uncle Sam, is good enough for us (cheers). I believe that if we follow the flag (loud cheers)—the flag, gentlemen, which, planted on the bloody field of Lexington, flutters in the breezes that waft the perfume of liberty to our island possessions beyond the sea—" (vociferous and prolonged cheering). Thereupon it becomes the sense of the meeting to adopt the Government rule of "Multiply one-sixth the tensile strength, etc.," for the establishment of pressure on steam boilers! All this may be patriotism, and it may get votes, but it is not common sense.

The present Government rule is so obselete, so incomplete, so inconsistent, that were it not also dangerous, it would be silly. It is so ridiculous that, as pointed out on page 874 in these columns for November 23, the Government itself is specifying that its boilers shall not be built under its own rules, while in his annual report to the secretary of Commerce and Labor for the year ending June 30, 1905, the supervising inspector general in speaking of the Government's factor of safety says: "It has long since passed the period of its usefulness, and is no longer adequate to meet the conditions it was designed to govern."

The rule is incomplete, because it does not take into consideration the efficiency of the longitudinal seam. If there is no longitudinal seam, the rule is all right and gives a factor of safety of 6; but there always *is* a seam, and if its efficiency is 70 per cent., the factor of safety is not 6, but 70 per cent. of 6, or 4.2, while if the seam has an efficiency of 55 per cent., the factor is but 3.3.

The superb fatuity of the rule seems to culminate in this—that if one finds that a double-riveted shell will not give the pressure wanted, and has the boiler built triple-riveted, he is not allowed one extra pound of pressure, nor if the shell is built with butt joints with double covering strips, and the whole triple riveted, or even quadruple riveted with its eight rows of rivets—nay, even with a most approved joint of 98 per cent. efficiency—one is not allowed an extra pound of pressure over that stipulated for the double-riveted lap seam.

To the natural question, why are such things allowed to be, the answer is that the ship owners who do not want the bill changed, block all efforts of Congress to bring this very desirable end about. It may be said that in the past the Government rule affected only those who chose to be so affected, but now, when States and municipalities are waking up to this question, the Government rule is having a pernicious influence on the whole country greater than ever before, and it is high time that the untiring efforts of those who have sought to have the law changed should bear fruit. We do not advise jamming through a rule which would tie up half the boats on the Mississippi river; let the old boats work out their own salvation, but adopt a safe and sane rule which shall apply only to new boilers built after the passage of the new rule.

As to the ease with which the present bad law could be changed to a new good law, all that would have to be done would be, roughly, to add three letters, namely, "net." Then the rule would refer to the strength of the net plate and not to the plate. And the inspectors would not have to worry about the boilermakers not putting in good joints, for then there would be the premium of greater allowable pressure, for good design. Then too, the apparent, or fancied factor of safety of six, could be changed for a real factor of 4½ or 5.

Under no circumstances should a stop valve be permitted between the boiler and its safety valve.

Efficiency Tests of Melville Gear

When the Melville and Macalpine experimental reduction gear was first erected and started on September 7, 1909, extensive preparations had been made to determine the efficiency of the apparatus on the assumption that there might be a transmission loss of at least 5 per cent. After calculating the results of a few preliminary trials, it was found that the apparent efficiency was over 98 per cent., and this raised the perplexing question as to how to determine this unexpectedly small transmission loss with a satisfying degree of exactitude.

According to H. E. Longwell, an endurance test at maximum load and speed was started at 3.15 p.m., on October 16, 1909, and continued until 7.15 a.m. on October 18, or a total of 40 hours. The load was applied by the hydraulic dynamometer described elsewhere in this issue. During this trial the following average conditions were maintained:

Net weight on scale, lb	4055.4
Speed, r.p.m.	300.6
Brake horsepower	6048

There was nothing in the operation of the gear to indicate that this load might not be carried indefinitely.

There being no way in which to measure the indicated horsepower of a steam turbine, it became necessary to establish the exact brake horsepower in some other way. There is one characteristic of the steam turbine that makes it possible to calibrate any particular machine, in such a way that its output in effective horsepower at any instant may be determined with greater accuracy than it is possible to determine even the indicated horsepower of a reciprocating engine. As long as the speed and exhaust pressure are maintained constant, the absolute inlet pressure of commercially dry steam, at any instant, is a very accurate measure of the brake horsepower the turbine is developing.

By substituting for the reduction gear a dynamometer connected directly to the turbine shaft, and operating the turbine at a fixed speed, and with a constant vacuum in the exhaust pipe, the inlet pressures corresponding to different loads at this speed may be determined.

Plotting these observations on a diagram in which the ordinates represent absolute pressures per square inch, and the abscissas represent brake horsepowers, and drawing a line through the several points, a scale is obtained by means of which the horsepower corresponding to any inlet pressure may easily be read.

Table 1 gives the observed absolute inlet pressures, loads on scale, and calculated brake horsepowers over a considerable range above and below 5000 horsepower. From these observations and calculations a calibration curve similar to that shown, was drawn to a large scale, so that small differences of loads and pressure could be read accurately. With the reduction gear connected to the turbine, a number of observations were made of the absolute inlet pressures corresponding to practically the same range of loads transmitted through the gear to the slow-speed hydraulic dynamometer.

TABLE 1.

Inlet Pressure, Absolute.	Gross Weight on Scale.	Dead Weight on Scale.	Net Weight on Scale.	Brake Horsepower.
109.6	3585	385	3200	3675
114.8	3755	385	3370	3880
119.3	3925	385	3540	4076
124.3	4095	385	3710	4271
128.8	4265	385	3880	4467
133.3	4435	385	4050	4663
139.3	4690	385	4305	4956
143.8	4860	385	4475	5152
147.3	5030	385	4645	5348
151.8	5200	385	4815	5543
157.8	5455	385	5070	5837

In these tests, as well as in the turbine calibrations, at every load before reading the inlet pressure, the speed of the turbine was brought exactly to 1500 revolutions per minute as indicated by a vibration tachometer, and the vacuum adjusted to exactly 22.78 inches corrected to a barometric pressure of 30 inches.

In Table 2, column 1, are the observed absolute inlet pressures corresponding to the different loads transmitted through the gear as set down in column 5. In column 6 are given the horsepowers of the turbine alone corresponding to the same inlet pressures, as determined from the calibration curve. Obviously, the quantities in column 5 represent the power output of the gear, and those in column 6 represent the power input. The quantity in column 5, divided by the quantity standing against it in column 6, gives the efficiency as it appears in column 7.

TABLE 2.

Inlet Pressure, Absolute.	Gross Weight on Scale.	Dead Weight on Scale.	Net Weight on Scale.	B.H.P. Delivered by Gear.	B.H.P. of Turbine from Calibration Curve.	Efficiency.
111.8	2900	388	2512	3712	3771	98.7
122.3	3200	388	2812	4156	4197	99.0
132.3	3480	388	3092	4576	4623	98.9
142.3	3791	388	3403	5036	5108	98.7
152.3	4100	388	3712	5486	5567	98.5
162.3	4400	388	4012	5927	6057	98.7

While the efficiencies as calculated from these observations are consistent and uniform over the whole range of observations, a check of the efficiency figures was made as follows:

The gear is lubricated by circulating a copious supply of oil under a head of about 10 pounds, through the bearings, and through a spray which plays continuously on the teeth of the wheels. The transmission loss in the gear, therefore, appears as heat in the oil. The heated oil coming from the gear is passed through a cooler which is very like a surface condenser with cold water circulating through the tubes. By measuring the quantity of oil circulated and noting the rise in temperature in passing through the gear, a close approximation of the number of B.t.u. lost per hour in friction is obtainable.

At a load of 5088 horsepower delivered by the gear, 591 pounds of oil were circulated per minute, with an average rise in temperature of 9.86 degrees Fahrenheit. The specific heat of this oil is 0.47, and consequently the total heat absorbed per hour is

$$591 \times 0.47 \times 60 = 164{,}208$$

B.t.u. As 2545 B.t.u. per hour is the equivalent of a horsepower, the total heat accounted for in the oil is

$$\frac{164{,}208}{2545} = 64.17 \textit{ brake horsepower.}$$

The brake horsepower being 5088, and 64.17 horsepower being accounted for by the heat in the oil, the efficiency by this method of calculation would be

$$\frac{5088}{5088 + 64.17} = 98.75 \textit{ per cent.}$$

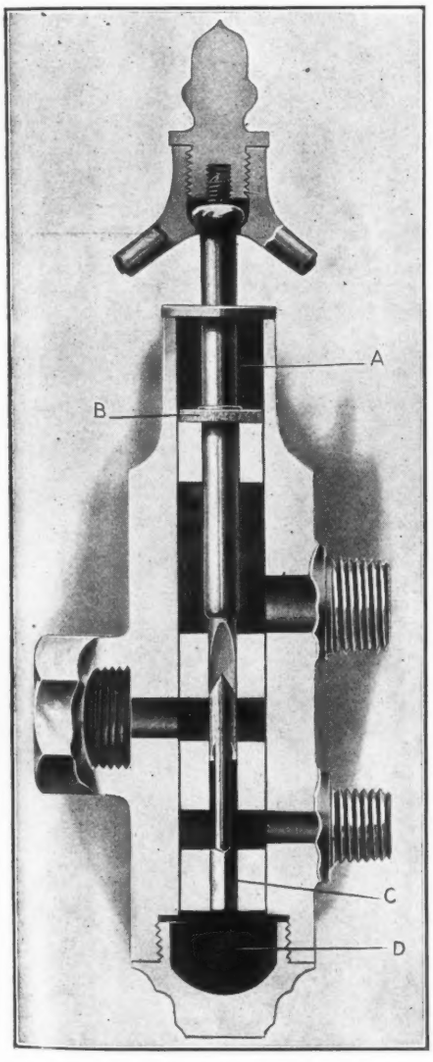
Giving due weight to both methods of measurement, it is quite evident that at approximately 5000 horsepower, with the turbine running at 1500 revolutions per minute, the efficiency of the gear is more than 98.5 per cent. Less elaborate and extensive determinations show an efficiency of 98 per cent.

NEW·POWER·HOUSE ▾ EQUIPMENT ▾

What the Inventor and the Manufacturer is doing to save time and money in the Engine Room and Power House

Improved Valve for Locke Regulator

The illustration shows the design of the improved water valve of this regulator, which is the most important part. Heretofore, the spindle of this valve



SECTIONAL VIEW OF IMPROVED VALVE

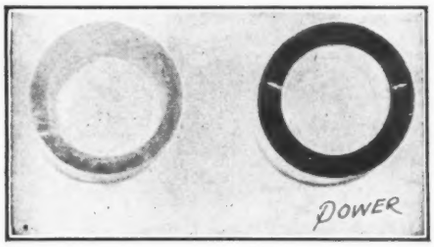
worked through closely fitting rings at both top and bottom of the valve, which was liable to become roughed up and stop the operation of the regulator. In the improved valve both are done away with. An oil cup *A* is formed about the spindle at the top, having a soft felt washer *B* at the bottom of the chamber which allows the spindle to work in oil, keeping it perfectly lubricated.

By dispensing with the closely fitting lower portion of the spindle *C*, and using it simply as a guide, another source of trouble is got rid of, as it was continually coating up with iron rust, causing it to stick, and when the valve became worn and leaky the water that escaped around the spindle went to waste. No leak can exist at this point now as the exhaust chamber *D* has the full pressure of water where it acts beneath the end of valve spindle *C*, greatly assisting in the operation of the regulator.

The improved valve is gotten out by the Locke Regulator Company, Salem, Mass.

A Clear Gage Glass

Around the steam plant many accidents are caused by the breaking of gage glasses. The flying particles of glass



"SONDERGLAS" SCOTCH GLASS

often cause a bad cut, or a bad scald may be obtained from the escaping hot water or steam. The importance, therefore, of using a gage glass designed for modern pressure is apparent. It is also important to have a gage glass that remains clear and does not become cloudy or obscured, so that the water level will be plainly visible at all times. These features are claimed for the new German compound water-gage glass, known as "Sonderglas," and supplied by A. W. Chesterton Company, 64 India street, Boston, Mass.

From the accompanying photograph it is evident that the Sonderglas is much clearer than the usual Scotch glass. The circle at the left is an end view of the Sonderglas, and the one at the right an end view of the Scotch glass commonly used. There is a considerable difference in the transparency of the two, and this is more evident when looking at the originals.

An attempt was made to show the difference in transparency of the two glasses by taking a photograph of them standing endwise in front of a printed page. Although there was some difference in the legibility of the type under the two glasses, this difference was not great enough to show in a reproduction of the photograph.

To allow for changes of temperature and for expansion and contraction the Sonderglas is composed of two tubes fused into one. It is claimed that this method of manufacture adds to the pressure-resisting qualities of the glass and produces a product which will be unaffected by alkalies or acids.

Angular Variation Indicator

The accompanying engravings illussrate the construction of an instrument for indicating variations in the rotational

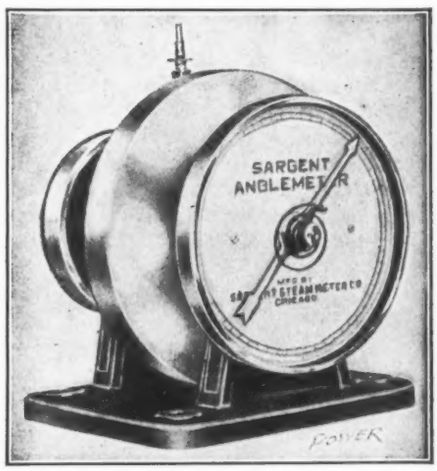


FIG. 1. ANGULAR-VARIATION INDICATOR

velocity of reciprocating engines or any other revolving machine that does not rotate at a uniform rate. The instrument was designed to meet the need of some simple means for determining the variations in angular velocity of reciprocating engines intended for driving alternators in parallel. It consists, broadly, of a heavy flywheel arranged to be driven by a pulley through a "floating" bevel-gear system, and a pointer which indicates differences between the speed rate of the pulley and that of the flywheel.

Fig. 1 is a front view, showing the dial, flywheel and one edge of the pulley; Fig.

2 shows the pulley end and the floating gear arrangement between the pulley and the flywheel. The sectional view, Fig. 3, gives a clear idea of the rotation between the various parts. The pulley *C*, which is of the step type with two diameters, drives the bevel gear *E* through the feather *I*, the hub of the gear being extended to form a hollow shaft for that

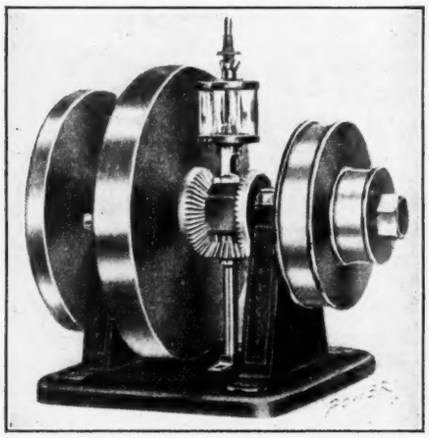


FIG. 2. REAR VIEW OF ANGLEMETER

purpose. The main shaft *A* carries a lateral stud *D* on which revolves the intermediate gear *H;* this gear transmits the motion of the pulley to the flywheel *B*, which has a bevel gear cut in its hub. The stud *D* is extended downward nearly to the base and its end plays between two stops on the base, one of which appears at *K*.

The pointer *O* is connected to the spindle *A* through a delicate multiplying gear consisting of two pinions *G* and *G'* and two segmental racks *F* and *F'*. Consequently, any oscillation of the spindle *A* is magnified 100 times at the needle *O*. A spiral spring *X* tends to hold the pointer

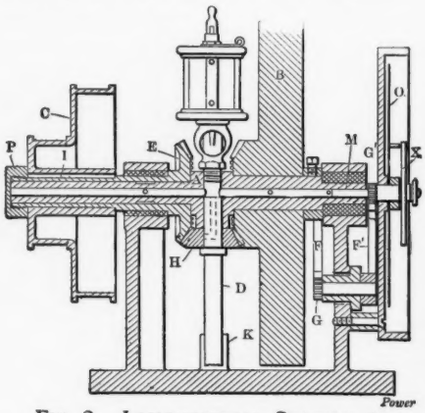


FIG. 3. LONGITUDINAL SECTION OF ANGLEMETER

at zero, and when it is in that position the end of the stud *D* is midway between the two stops on the base.

So long as the speed of the pulley is constant, the spring *X* will hold the pointer at zero. If the pulley speed should decrease momentarily, however, the flywheel *B* will continue to revolve at the old speed and the dropping behind it of the pulley speed will cause the bevel gear *H* to swing the stud *D* out of its central position; this motion will be indicated, of course, by the pointer, and magnified 100 times.

The instrument is driven by an endless belt of special cotton tape which is practically inelastic and therefore transmits speed variations of the prime mover without modifications. The two pulley diameters are provided in order to apply the instrument to prime movers of widely different speeds; one is 6 inches and the other 3 inches in diameter. When the instrument is driven by an engine pulley 7.2 times the diameter of its own pulley, the pointer indicates angular variations directly; each large division of the scale on the dial then represents one minute and each small division 10 seconds of angular "swing" at the engine shaft.

The instrument is manufactured by the Sargent Steam Meter Company, Chicago, and is known as the Sargent indicating anglemeter.

Twin Vertical Air Pump and Jet Condenser

The twin air pump and condenser, shown in front and back views, Figs. 1 and 2, is manufactured by the Blake Pump and Condenser Company, Fitchburg, Mass.

It consists of twin vertical steam cylinders, when made to run noncondensing, and compound steam cylinders when designed to run condensing. The cylinders are, on the pump illustrated, 18x24 inches, the water cylinders being 34 inches in diameter and are double acting. They are constructed with Tobin bronze lining, and the water pistons are of composition metal. The same may be said of the suction and discharge valve decks with which valve seats are cast solid, although cast-iron decks may be used by screwing in composition valve seats in the usual way, when cheaper construction is required.

The water piston, head packing and water valves are made easy of access, through a handhole on each side of the pump. The valves are held in place by saucer-shaped guards and Tobin bronze studs with lock nuts, and are operated without the aid of springs. Tobin bronze is used for the water piston rods, while steel is used on the steam piston rods.

In Fig. 1 is shown the valve gear and general design of the rocking beam, in which crossheads are fitted at each end, forming a Scotch yoke for the composition slide blocks to work in. The control-

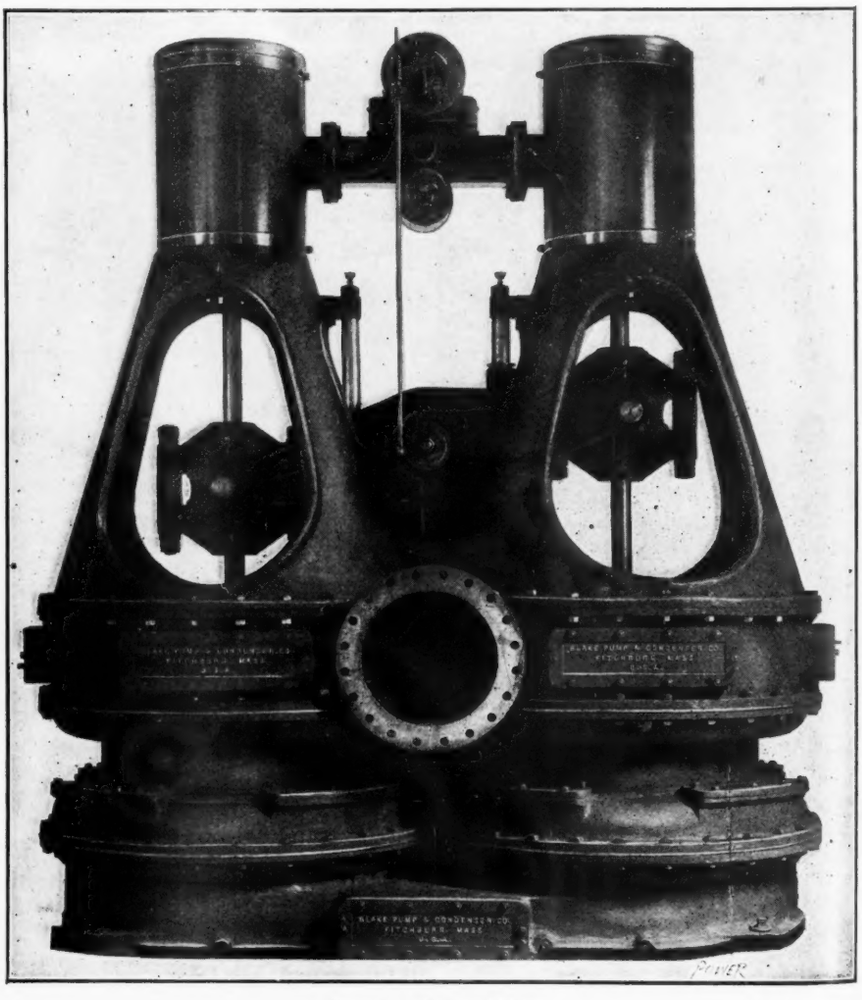


FIG. 1. FRONT VIEW

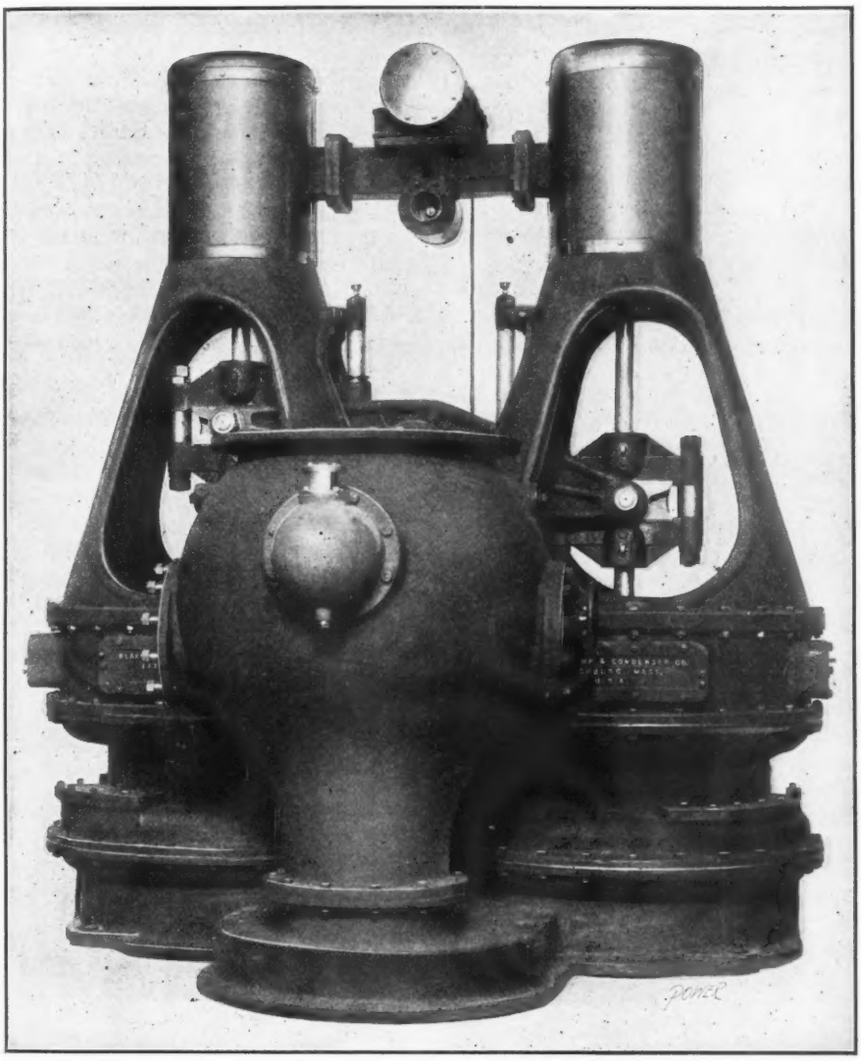


FIG. 2. BACK VIEW

ling valve is of an improved steam-actuated type, on which patents are pending.

The location of the condenser valve is shown in Fig. 2. The injection rod is made of Tobin bronze, the cone being of composition metal, the same being operated by bevel gears having a ratio of 2 to 1. They are accessible through flange openings below and the vacuum breaker opening above.

The amount of water passing through the condenser is regulated by the vertical adjustment of the injection cone, which acts as a nozzle to form a thin spray, which is thrown out at an angle of 45 degrees. As this water falls on a shelf it forms a secondary spray through which the steam from the engine must pass.

The injection and water of condensation flow through the bottom of the condenser by gravity into the pump.

To prevent flooding of the engine, a new design of independent balance-valve vacuum-breaker attachment is secured to the side of the condenser. When the water reaches the level of the float chamber, a float raises the check valve, which allows an inrush of air, which breaks the vacuum and prevents further drawing of water into the condenser.

The particular unit from which the photographs were taken was recently shipped to the Boston Elevated Railroad Company to be used in connection with one of their 5000-horsepower vertical-type engines.

Westinghouse Three-Phase Transformer

Fig. 1 herewith shows the "working" part of a three-phase transformer built by the Westinghouse Electric and Manu-

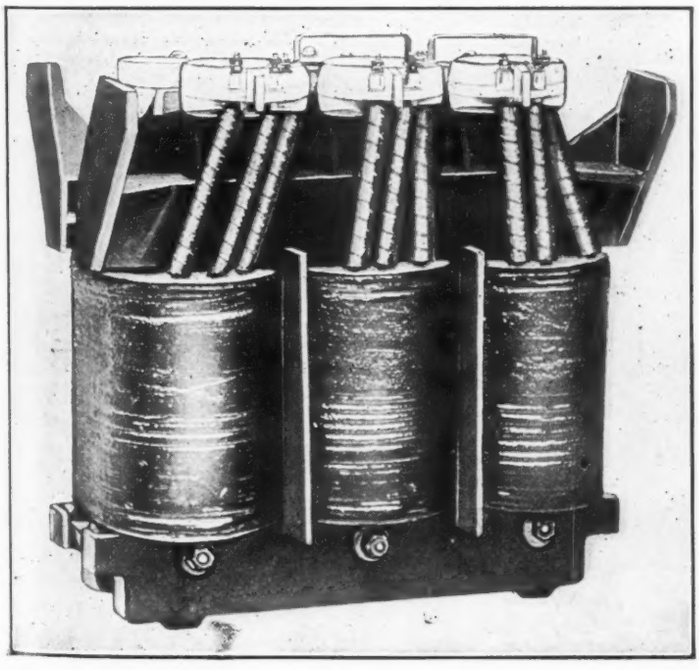


FIG. 1. INTERIOR STRUCTURE OF THREE-PHASE TRANSFORMER

FIG. 2. EXTERIOR OF THREE-PHASE TRANSFORMER

facturing Company, Pittsburg, Penn. The transformer consists of three cores or legs joined by a common yoke piece at the top and bottom and each carrying the primary and secondary windings for one phase of the three-phase circuit. In other words, each coil (comprising one primary and one secondary winding), with its core and share of the yokes, is the equivalent of one of the transformers used in the familiar triple-transformer arrangement.

The three-phase transformer is much lighter in weight than three equivalent single-phase transformers and its losses are less, because of the fact that each two cores form a return magnetic circuit for the third core, and therefore there is much less active iron required than for three separate transformers.

Fig. 2 shows the external appearance of the transformer, which is not greatly different from that of a single-phase transformer. The case is of corrugated cast iron like that of the Westinghouse Type S single-phase transformer. The three high-tension leads enter the case on the front side, away from the pole to which the case is attached, and the low-tension leads enter on the opposite or pole side. Separate terminal blocks are provided for the high-tension and low-tension leads of each of the three phases. The low-tension winding on each leg is divided into two parts which may be connected in series or in parallel, as is commonly done with single-phase transformers.

Demon 2-inch Tube Cleaner

A new form of cleaner for small tubes is herewith illustrated, Fig. 1. The principal obstacle in the way of the manufacturers of mechanical tube cleaners has been the problem of designing a motor small enough to enter the tube and yet which would develop an abundance of power to remove the scale in a reasonable time.

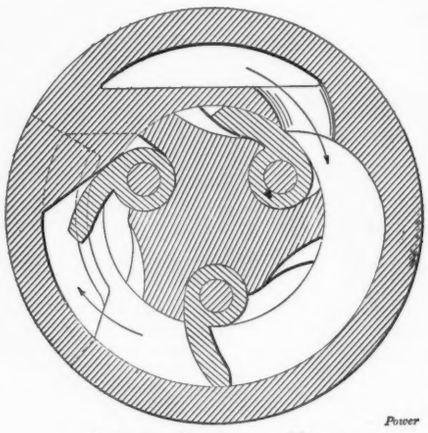


FIG. 2. SECTIONAL VIEW

Referring to the sectional view, Fig. 2, it is seen that the hydrostatic pressure is used directly upon rotary pistons, which are arranged lengthwise or parallel with the tube so a comparatively large piston area can be secured.

The motor consists primarily of a body and a main shaft located centrally. Hinged longitudinally with the shafts are three pistons, which are so arranged that the water automatically throws each one in turn in its working position and each piston folds in when the water has rotated the piston to the exhaust port. The result is that this small cleaner develops sufficient torque or twisting power to remove quickly and thoroughly any incrustation which may be found in these tubes. The different parts of the motor 2-inch cleaner is shown in Fig. 3.

In order to produce a uniform action of the cutter head on all sides of the tube the cleaner is held centrally in the tube by centering lugs. The cutters are mounted on two arms or shafts which are pressed tightly against the scale by stiff springs within the cutter-head body. In this way the cutters are forced to cut and cannot slide over and pass by the scale.

The cleaner is manufactured by the General Specialty Company, 887 Niagara street, Buffalo, N. Y.

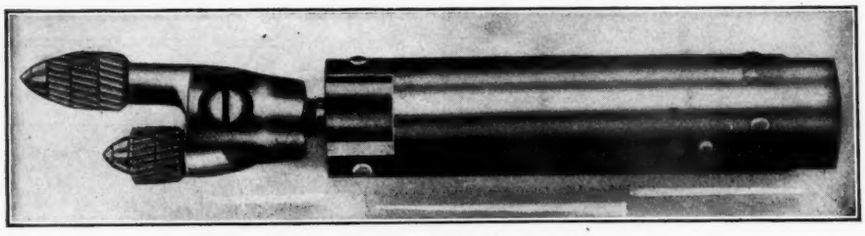
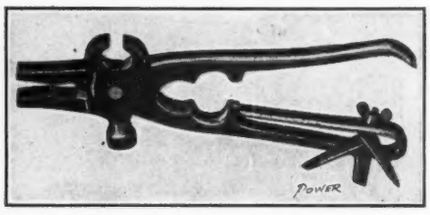


FIG. 1. TWO-INCH TUBE CLEANER

FIG. 3. PARTS OF MOTOR

A Combination Tool

The combination tool, shown in the illustration, weighs 2½ pounds and consists of wire pliers, with side jaw wire nippers, alligator wrench, hammer, nippers, gas-pipe wrench, riveter, vise, screwdrivers, reaming awl, tack and nail puller; a whole kit of tools in one.



COMBINATION TOOL

It is made by the Mansfield Machinery and Supply Company, Mansfield, O.

The only thing about perpetual motion which cannot be stopped is the attempt to invent it. A. T. Markwood, of Johnson City, Tenn., is the latest wizard who has tapped the rock of imperturbable nature and made it yield power for nothing. According to the *Chattanooga Times*, the principles on which this machine is to be made to go (note the future tense), have been chasing through the brain of the inventor and now they are being put into visible shape. In running this new machine neither coal nor fuel of any kind will be required, but like a mighty giant it will have its own power. You have only to touch a button and the machine moves, until by a magic press on the button the invisible spirit of motion is made to cease.

INQUIRIES OF GENERAL INTEREST

Questions are not answered unless they are accompanied by the name and address of the inquirer

An Idler with a Riding Belt

We have a 22-inch double belt, with a 16-inch double rider, transmitting from 180 to 190 horsepower and arranged as in the accompanying sketch. When the idler is raised into contact with the belt, the 16-inch rider runs off. Should the idler pulley be convex, concave or flat face, with or without flanges?

C. L. C.

The idler may be either flat or convex, but must not be concave and must not have flanges. The essential thing is that its axis shall be perfectly parallel with that of the pulley onto which the belt is leaning. If the axis of the idler is not directly at right angles to the intended movement of the belt, it will lead the belt off just as a skewed roller will lead a plank off on the side toward which it is turned.

Formulas for Piston-rod Sizes

What should be the size of a piston rod for a cylinder 15 inches inside diameter to withstand a working pressure of 2400 pounds per square inch?

M. D.

Let

D = Diameter of cylinder in inches,
p = Greatest unbalanced pressure on piston in pounds per square inch,
L = Length of rod in feet,
d = Diameter of rod in inches.

Unwin gives for steel rods

$$d = 0.0144\, D \sqrt{p}.$$

Thurston gives for steel rods

$$d = \sqrt[4]{\frac{D^2 p L^2}{a}} + \frac{D}{80}$$

a being from 10,000 to 15,000 according to conditions of shock, etc.

Marks:

$$0.0105\, D \sqrt{p}$$

and

$$0.03525 \sqrt[4]{D^2 l^2 p}$$

in which l is the length of the rod in inches. Of Marks' formulas, try the first, and if the diameter is less than 1/12 of the length, use the latter.

Steam Required to Heat Water

I have to estimate the amount of power consumed by each department of our plant, as we use steam for boiling liquids and heating water. In some cases the steam is blown direct into the water and in others the steam flows through a coil and the condensation is trapped. In one tank, 26x35x15 inches deep, which was full of water at 73 degrees Fahrenheit, steam was turned on and in twelve minutes the temperature rose to 170 degrees Fahrenheit. I estimate that there was 493.75 pounds of water in the tank. How many B.t.u. were consumed and what is the equivalent horsepower?

F. C. M.

Your tank would contain

$$\frac{26 \times 35 \times 15}{1728} = 7.9 \text{ cubic feet.}$$

One cubic foot of water at 73 degrees Fahrenheit weighs 62.29 pounds. There are then in the tank

$$7.9 \times 62.29 = 492 \text{ pounds of water.}$$

To raise a pound of water from 32 to 170 degrees, takes 138.45 B.t.u.; from 32 to 73 degrees, takes 41.02 B.t.u. The difference, or from 73 to 170 degrees, would take 97.43 B.t.u.

Then to heat the tank of water from 73 to 170 degrees would take

$$492 \times 97.43 = 47{,}935.56 \text{ B.t.u.}$$

A boiler horsepower is equivalent to 33,305 B.t.u. per hour, or

$$\frac{33{,}305 \times 12}{60} = 6661 \text{ B.t.u.}$$

in twelve minutes. Then the boiler horsepower required to heat the tank is

$$47{,}935.56 \div 6661 = 7.2 \text{ horsepower.}$$

This charges only for the heat units delivered to the tank, and would be right if the water in it were heated by a coil and the condensation returned to the boiler without loss of heat except to the tank. It may be corrected by charging additional units that the boiler does not get back. If the steam mingles with the water you must know its pressure and quality. Assume it to be dry-saturated at 100 pounds absolute. Then each pound of it contains 1181.8 B.t.u. above 32 degrees, and

$$1181.8 - 138.45 = 1043.35 \text{ B.t.u.}$$

above 170. It would take

$$47{,}935.56 \div 1043.35 = 45.95$$

pounds of steam to furnish the heat by condensing to water at 170 degrees. If the feed water is at 73 degrees, it takes

$$1181.8 - 41.02 = 1140.78 \text{ B.t.u.}$$

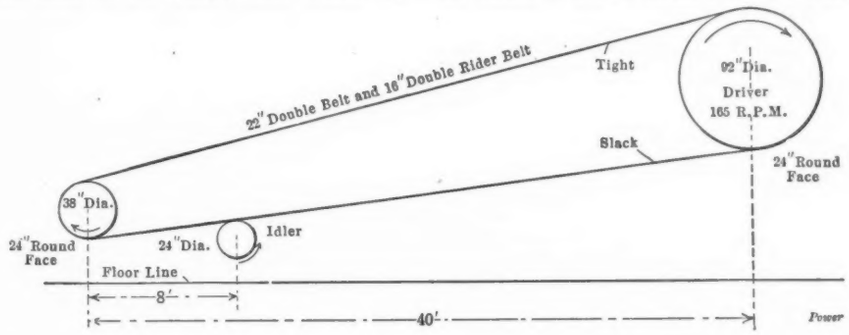
to make one pound of the steam, and

$$1139.98 \times 45.95 = 52{,}382.081 \text{ B.t.u.}$$

to make enough to heat the water. Dividing this by 6661, as before, shows that

$$52{,}382.081 \div 6661 = 7.86$$

horsepower must be expended for 12 minutes under the assumed conditions to heat the water.



Preventing Smoke

Is there any device on the market which effectively prevents smoke from bituminous coal used under boilers?

L. B. B.

The smokeless combustion of bituminous coal is more a matter of furnace proportion and operation than of any patented device, although a method of continuously introducing the fuel, without the necessity of opening fire doors, is a great help. Bituminous coal can be burned at a reasonable rate, without the creation of offensive smoke, except at the time of cleaning fires. Write to the University of Illinois, Engineering Equipment Station, for their bulletin, No. 15, on "How to Burn Illinois Coal without Smoke," by L. C. Breckenridge; also to the United States Geological Survey for a copy of D. T. Randall's report upon the "Burning of Coal without Smoke in Boiler Plants." The number of the bulletin is 334,228.

Book Reviews

POCKET BOOK OF ELECTRIC LIGHTING AND HEATING. By S. F. Walker. Published by The Norman W. Henley Publishing Company, New York. Leatherette; 438 pages, 4x6½ inches; 272 illustrations; 246 tables. Price, $3.

In the preface the author says he has "not hesitated to go right away from the beaten track of other Pocket Books" in preparing this one. We cheerfully confirm his statement. He has gone "right away from" every sensible precedent with which we are familiar, and has produced a collection of catalog data interspersed with feeble rudimentary attempts to present principles of operation and construction. As a sort of catalog synopsis, the book is fairly commendable; as a pocket reference book for engineering data it is pitifully inadequate. The author's precision of scientific statement may be judged by these specimens: "The dynamotor is a motor-generator, only applicable to the transformation of continuous-current machines." A transformer "is built up of a number of laminated iron plates in such a form that two separate windings may be accommodated within the magnetic field of one of them."

The section devoted to cables is excellent as a compilation of practical data, and far superior to the remainder of the book, from any viewpoint.

ENERGY. By Sidney A. Reeve. Published by the McGraw-Hill Book Company, New York; 1909. Cloth; 240 pages, 6x9 inches; illustrated. Price, $2.

To look at the universe with a comprehensive vision that comprises regions to which the telescope and the microscope do not reach; to apply the laws of celestial mechanics to molecular physics, to trace the orbit of an atom in the terms in which an astronomer traces the orbit of a world and in doing this to bring out an adequate conception of the nature of energy in all its changing scope from the state in which it is nearly all movement and little separation to that in which the separation factor becomes predominant at the expense of motion, is the object of this work.

Energy is commonly defined as force in motion, the product of force and space, and heat, which is but a form of energy, as "a mode of motion." Professor Reeve's book, for the author is a past professor at the Worcester Polytechnic and is the writer of a well known work on thermodynamics, expands this expression, with which the usual teacher of and writer upon energetics has been contented to stop. He shows what mode of motion is this heat to which all forms of energy tend, and in doing so clears away the necessity of the conceptions of the unattainable and impossible absolute zeros of pressure temperature and motion and of "perfect elasticity" with which the science of molecular physics has been lumbered.

There can be no such thing as absolute motion. To a man aboard ship the objects about him are stationary but they are changing their latitude and longitude with the motion of the vessel. A building is stationary upon the earth, but it is moving with a velocity measured in miles per second with reference to the sun. Everywhere is motion. Absolute fixity or rest is unknown and inconceivable, but it is relative. All force is relative. It exists only *between* bodies and may never be imagined as exerted absolutely independently of mass. Energy must be considered as existing *between* masses. When a pendulum is in its highest position all its energy is that of position, of separation between its mass and that of the earth. When it is in its lowest position its energy is all that of motion. The idea of friction may be gotten rid of by imagining a spherical mass with a hole through it, and a smaller mass released from a point outside the larger mass and passing in a straight line through the hole and out the other side. At first its energy would be all that of position. As it gained momentum under the attraction of the larger mass the energy of position lost by approach would be converted into kinetic energy in the momentum of the moving mass. This would be greatest when the centers of the masses coincided, and there was no separation, just as in the case of the pendulum at the bottom of its swing; and as the smaller mass passed with this momentum beyond the center and out to a distance upon the other side equal to that from which it first fell the energy would again be converted from that of velocity to that of position.

But in nature there is no such thing as straight-line motion. We are accustomed to think of the straight line as the natural path of a moving body and to require some cause for its divergence therefrom, just as we are accustomed to regard rest as the normal state of matter and to wonder what has put it into motion. In fact rest is an exceedingly complicated condition requiring an absolute balance of forces in all directions and the chances are infinitely against an unrestrained body moving in a straight line. All masses are under the attraction and influence of other masses. One mass attracted to another will swing around it in a circle only if the eccentricity of its orbit be zero, and will approach it in a straight line only if the eccentricity of its orbit is infinite. If the eccentricity is unity, the path will be a parabola; but outside of these improbable and exceptional conditions the path will be an ellipse when the eccentricity is less than unity, and an hyperbola when it is greater than unity. There is no necessity for the assumption of the perfectly elastic atom or molecule. The molecular bodies do not impact but turn about each other in an orderly way.

All of this has been known of the astronomer for years, perhaps of the delver into molecular physics, but it has remained for Professor Reeves to amplify our conception of molecular interaction in the light of the application of the laws of Kepler and of Newton, and to do it so that other than a specialist can understand. His treatment of this abstruse subject is so plain and nonmathematical that, as he says, a knowledge of Kepler's and Newton's laws of motion and force, a little analytical geometry and enough of the concepts of the calculus to permit thinking of millions of molecules at once without becoming confused as to what they may or may not do is all that is required. And even he who lacks this equipment will be able to follow the main story and to glean from this presentation new ideas as to the ultimate nature of energy and the manner and meaning of its transformations.

The Buying and Handling of Steam Coal

This is the title of a pamphlet of 48 pages dealing with the above subject which has just been published and contains the report of the committee on fuel supply of the Boston Chamber of Commerce. In this report the committee have considered the fuel supply of New England, its source, quality and quantity, as well as questions affecting its cheaper production and means for economical consumption.

The report gives a comprehensive survey of the rate charges, with comparison graphically shown by maps, table of analysis of coal from the district supplying New England, and a good deal of other information.

The committee should be highly complimented on the work they have done, and especially in the production of the maps, which will be of much value to those interested in the coal question. One, the "All Rail" map, shows all-rail rates of bituminous coal by a color scheme and figures, with a key to the same, showing the maximum and minimum rate charges. There is also a table on the map which gives the additional charges to be added to the map rates when shipping coal over certain railroads.

The map "Tidewater" shows the rate